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Design and Evaluation of Low-Cost Stainless Steel—Fiberglass-Foam Blades for Large Wind-Driven Generating Systems

DEPARTMENT OF DEFENSE

PLASTICS TECHNICAL EVALUATION CENTER

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October 1982

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SUMMARY

This report describes the development of a low cost wind
turbine blade design based on a stainless steel/fiberglass
foam design concept and to evaluate its principle character-
istics, its low cost features, and its advantages and disad-
vantages. A blade structure was designed and construction
methods and materials were selected. Complete blade tooling
concepts, various technical and economic analysis, and
evaluations of the blade design were performed. A study was
made to determine the applicability of the blade design
concept to various blade lengths in the range of 15 ft. to
200 ft. A test specimen of the spar assembly, including the
root end attachment, was fabricated. This 20 ft. long full-
scale section will be fatigue tested by NASA.

It was concluded that the stainless steel/fiberglass-foam
blade concept is a viable design for application to large
wind turbine systems. A blade design was completed which
meets the requirements for operation on the MOD-OA wind
turbine. The design concept is applicable to various length
blades. Blades less than 60 feet would use a stainless steel
spar and blades of larger size would have a fusion welded
carbon steel spar. The design permits a choice of a wide
selectivity of air-foil shapes and, because of its modular
construction, the design provides producibility and quality
in high volume production.

INTRODUCTION

Studies conducted by The Budd Company Technical Center
resulted in a blade concept that NASA-Lewis Research Center
felt, held promise for a practical low cost wind turbine
blade.

As a result of the Federal Government efforts to encourage
development of alternate energy generating systems, the
NASA-Lewis Research Center, acting for D.O.E., awarded con-
tract DEN-3-129 to The Budd Company Technical Center. The
contract specified the requirements for designing and eval-
uating a low cost blade for large wind driven generating
systems.

The blade is composed of three structural elements:

A main box spar with root connection manufactured from
stainless or carbon steel, leading and trailing edge modules
of fiberglass reinforced plastic fabricated from low cost
commercial grade material, stiffened by rigid urethane foam,
and a fiberglass tip extension similar to the construction
used on the leading and trailing edge modules. The fiber-
glass elements provide the aerodynamic surfaces.

This report covers the development of the blade design.

The purpose of the work performed under this contract is
to demonstrate that the blade design meets the contract
low cost objectives and the performance specifications.

DESIGN REQUIREMENTS

To provide understanding of the basic design requirements, a condensed version of the NASA-Lewis contracted specifications as well as modifications are presented as follows:

A. Blade Geometry

1. The blades shall be designed using the Airfoil Configuration NASA-230XX.

2. The blade shall have a linear twist of ten degrees from the blade root (32" from rotor center) to tip.

3. The maximum thickness of the airfoil section shall be 40% of the chord at the root and 18% of the chord at the tip and shall vary linearly between these stations.

4. The hub-to-blade flange shall be at 31.75 inch blade station. The flange at this location shall be such that the blade is capable of interfacing with the hub spindle on the Mod-OA wind turbine.

5. Acceptable blade planforms are shown in Figure 1. This figure shows a root cut-out of 25%. This is the maximum allowable cut-out and the contractor shall determine the cut-out for his design since it is desirable for performance to minimize root cut-out. Figure 2 shows the effect of root cut-out on yearly energy capture. In terms of cost tradeoff, one megawatt hour is worth approximately $100 of blade cost.

6. The blade length shall be 60 feet from hub flange to blade tip.

7. The objective of the proposed blade design shall be to minimize blade cost while achieving adequate performance. To achieve this, the following goals should be considered consistent with other requirements and manufacturing practicality:

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a. Minimize the thickness-to-chord ratio

b. Minimize the solidity ratio of the blades.
    (Where solidity is the area of the blades
    divided by the area of the circle swept by
    the rotor).

B. Operational Considerations

1. The blade shall be designed as part of a two-bladed
    rotor.

2. The blade shall be designed to operate at ambient
    temperatures -30°F to 120°F (-35°C to 49°C).

3. The blade shall not be susceptible to damage by
    sunlight, high humidity, salt spray or fungus.

C. Blade Design Loads

1. The Contractor shall design the blade to withstand
    the following two independent load conditions:

    a. A maximum flat wise wind load of 50 pounds per
        square foot on planform area.

    b. Fatigue bending moment distributions, both
        flatwise and edgewise, as shown in Figures
        3 and 4. The cyclic edgewise load distribu-
        tion shown in Figure 3 shall be scaled by the
        ratio of the proposed blade weight to a blade
        weight of 2350 pounds (i.e., if the proposed
        blade weight is 3525 pounds, the cyclic edge-
        wise moment specification shall be 1.5 that
        shown in Figure 3).

2. Under these load conditions there shall be:

    a. No yielding of any structure under maximum
        wind load.

    b. No buckling of primary structure under maximum
        wind load.

    c. No failure in 4×10^8 cycles (30 years) under
        flatwise or edgewise fatigue loads. (Flatwise
        and edgewise positive directions are shown in
        Figure 5).

3. The fundamental first bending mode cantilever natural frequency (static) of the blade in the flatwise and edgewise directions shall fall in one of the following ranges:

$$1.50 \text{ Hz} \leq \omega \leq 1.85 \text{ Hz}$$

$$2.15 \text{ Hz} \leq \omega \leq 2.55 \text{ Hz}$$

$$2.75 \text{ Hz} \leq \omega \leq 3.15 \text{ Hz}$$

$$3.45 \text{ Hz} \leq \omega \leq 3.85 \text{ Hz}$$

Where ω = blade natural frequency

Note: It is permissable for the flatwise and edgewise natural frequencies to be the same.

4. Figure 6 shows an envelope of blade weight and cost. (The blade costs are those estimated as the average cost of 100 blades manufactured in one year). For a blade to be acceptable, it must fall within this envelope. Blade costs are to include all elements such as profit and amortization of tooling and engineering but shall be based on 1978 dollars (inflation shall not be considered). All blade weights are to include the weight of any flange adapter, if required. The lower cost-weight factor for a given design, the more attractive that design is.

5. The tolerance on blade weight for blade-to-blade uniformity is ± 2%. Spanwise center-of-gravity shall be on, or in front of, the 32% chordline and blade-to-blade variation of ± 1% of the chord length will be permissable.

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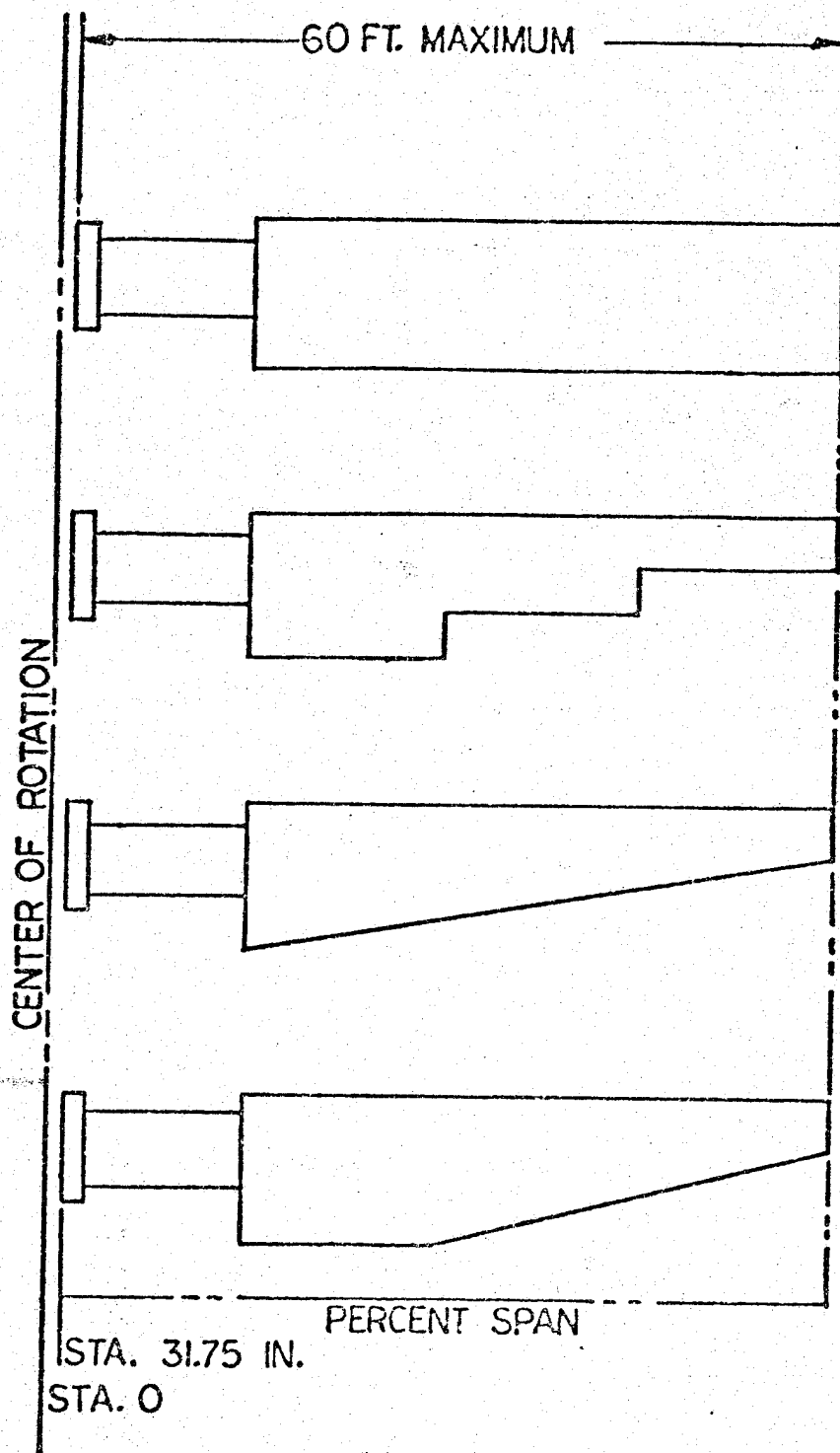
FIGURE 1 - ACCEPTABLE BLADE PLANFORMS

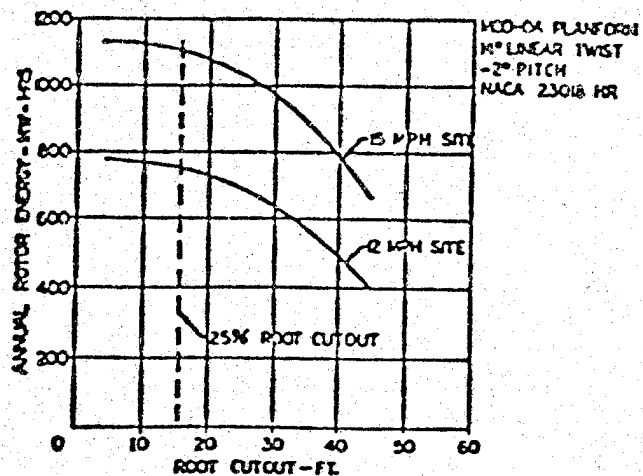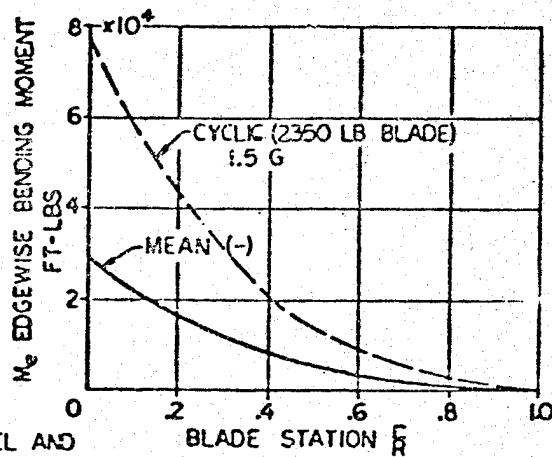FIGURE 2 - EFFECT OF ROOT CUTOUT ON ANNUAL ENERGY

NOTE: THE MOMENT VECTORS ARE PARALLEL AND
PERPENDICULAR TO THE .75 CHORD LINE
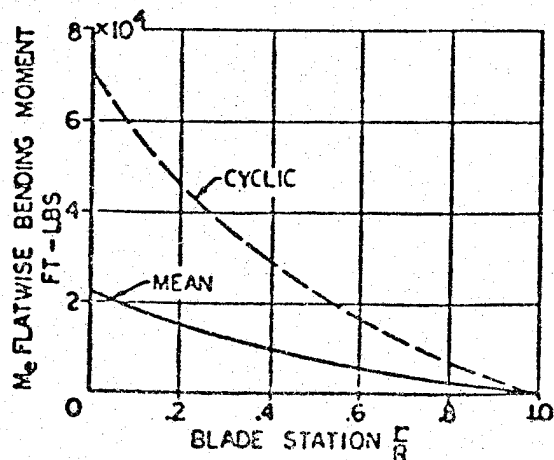
FIGURE 3 - FLATWISE & EDGEWISE BENDING MOMENT
DISTRIBUTIONS
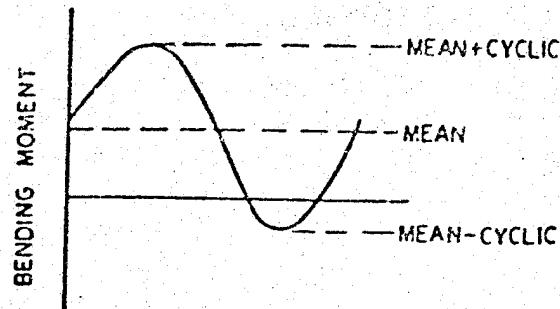
FIGURE 4 - DEFINITION OF MEAN AND CYCLIC MOMENTS

$+M_f$ - COMPRESSION IN HIGH PRESSURE SIDE
$+M_e$ - COMPRESSION IN LEADING EDGE
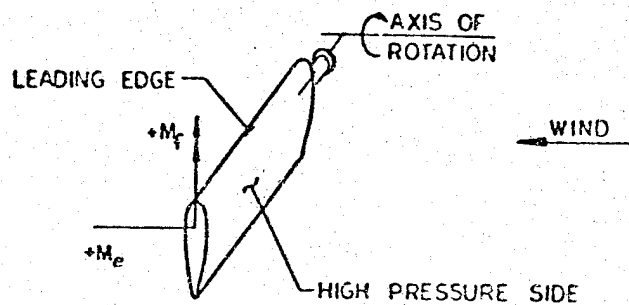
FIGURE 5 - BENDING MOMENT SIGNS CONVENTIONS
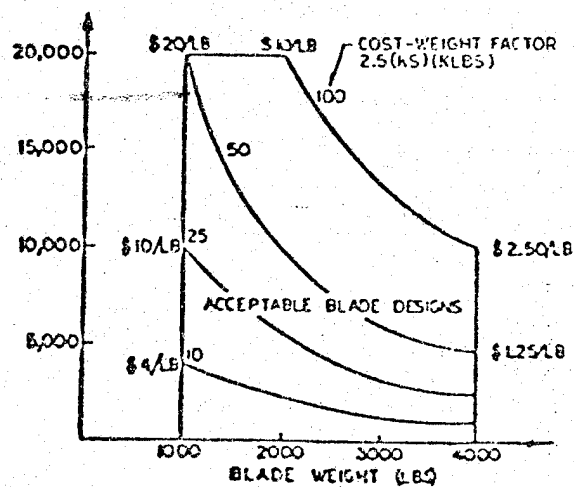
FIGURE 6 - BLADE WEIGHT VERSUS BLADE COST

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D. Supplementary Design Requirements

Clarification of blade design requirements was discussed during the contract initiation meeting held at The Budd Company in Fort Washington, PA. The Government clarified that a 47,000 ft.-lb. maximum static bending moment and a 3,000 lb. max blade weight would be required to provide compatibility with the MOD-OA wind turbine operational requirements.

BLADE DESIGN

In certain designs, the leading edge section, or the D
spar area of the blade, is used to carry the operating
loads, and the trailing edge is essentially non-structural,
carrying air loads for the trailing edge only. The Budd
design does not have a conventional forward D spar. The
design uses a central spine spar located within the envelope
of the aerodynamic contours. This spar carries all the
edgewise and flatwise loading and provides all the torsional
stiffness for the blade system.

The leading edge and trailing edge fiberglass components
are designed to distribute the air loads directly to the
spar and are segmented span-wise to prevent them from carry-
ing spanwise bending loads due to the spar deflections.
These sections are bonded to the spar using an elastomeric
adhesive. Figure 7 shows an idealized cross-section of the
blade.
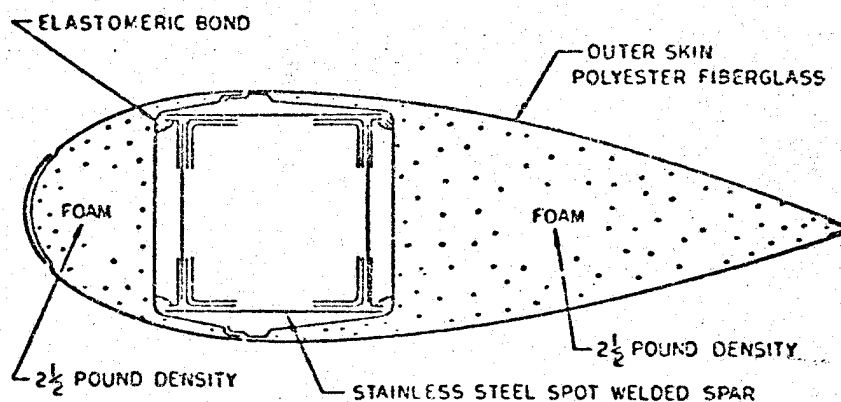
FIGURE 7 - TYPICAL SECTION WIND TURBINE BLADE

There is a center spar composed of spot welded stainless
steel. This structure is composed of top and bottom cap
strips, two shear webs, one on the front and one on the
aft side of the spar. The spar is built with a 10° twist
from the root end to the outer end. The leading edge and
the trailing edge assemblies are fabricated of fiberglass
reinforced plastic composed of multiple pieces bonded together
that are then filled with urethane foam. These fiberglass

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subassemblies are bonded to the spar at the four flange
corners of the spar. The leading and trailing edge assem-
blies are also bonded and mechanically fastened to each
other at the high camber point of the blade. The leading
edge of the blade is protected by an elastomeric cover sheet
to provide protection against impact of hail and other abra-
sive elements. The leading and trailing elements are designed
so as not to contribute significantly to the structural stiff-
ness of the blade.

Shown in Figure 8 is an exploded view of the spar assembly.
The spar is composed of four sub-assemblies, an upper and
lower cap strip plate assemblies and front and rear spar
web assemblies. The selection of stainless steel and the
spot weld process provides a unique method by which the
spar stiffness can be tapered effectively to provide a near
uniform stress from the root end to the tip of the spar.
This tapering is accomplished by the use of tapered angles
that are spot welded together and are joined to the upper
and lower cap strip plates. The thickness of the top plates
are tapered in three steps using a butt arc weld to join each
thickness. This is a specialized process that was developed
during the Budd 301 testing program which is discussed in
the section on the Test Program. This provides a weld of
high reliability in fatigue strength. By controlling this
taper, we are able to provide uniform tapering of the
basic properties of the spar section. The angles are first
tapered in the blank and, as a result, there is essentially
no material lost. The angles are then formed and then spot
welded to the spar cap assembly. This is all done in the
flat and then they are elastically twisted to match the 10°
twist of the spar. The spar webs are composed of two angles
and spar web. The two angles and web are spot welded in
place to form the web for the spar. There are two of these
subassemblies which are also built flat and elastically
twisted to form the 10° twist to the spar. The four sub-
assemblies are then inserted into an assembly fixture which
provides the 10° twist, and are then spot welded together.
This provides a very efficient tapering of the spar without
machining and at a very low cost using rolled sheet material.
The use of stainless steel also provides excellent corrosion
protection for long life of the spar. Spot welding is a low
cost, reliable, efficient attachment process to assemble the
spar.

Figure 9 shows a breakout of all the major subassemblies of
the blade.

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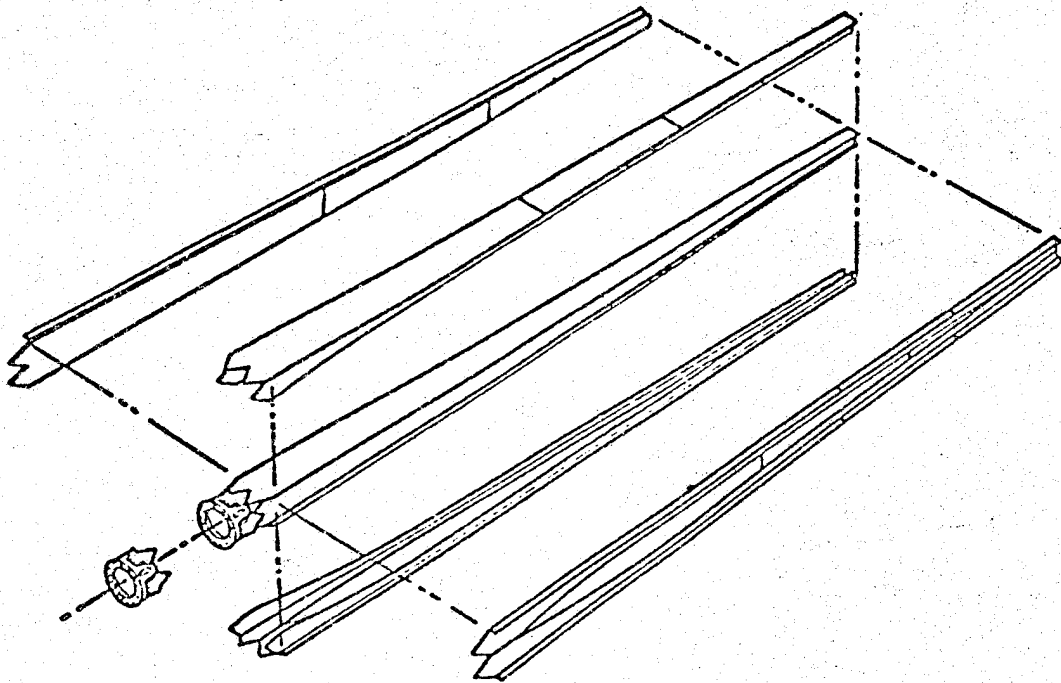
FIGURE 8 - EXPLODED SPAR ASSEMBLY
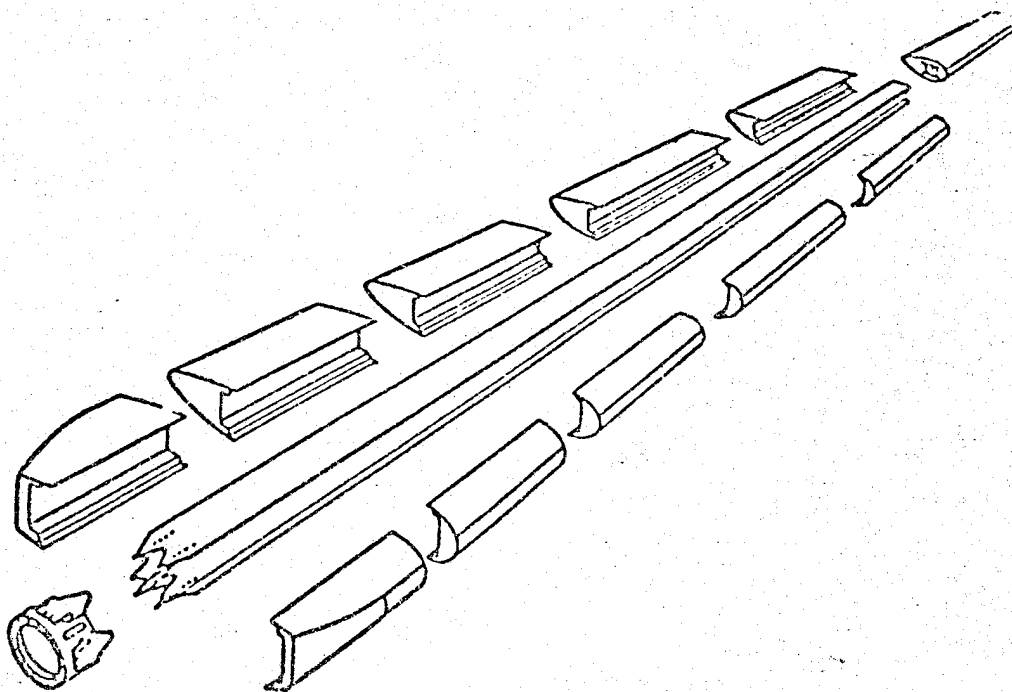
FIGURE 9 - MAJOR SUB-ASSEMBLY

Figure 10 shows an exploded view of the leading edge and
trailing edge assemblies. These assemblies are composed
of low cost layup of fiberglass-reinforced polyester.
Fiberglass elements are parasitic to the primary spar
structure and are used only to distribute the air loads
to the spar. Their design requirements are minimal. To
stabilize these elements for fatigue, the fiberglass
elements are filled with a semi-rigid, urethane foam of
approximately 2-1/2 pounds per cubic foot density. This
provides a lightweight, well-damped structure for the
leading and trailing edge assemblies and prevents aero-
dynamic flutter of the lightweight surfaces. This design
permits accurate dimensional control of the aerodynamic
surfaces at low cost.
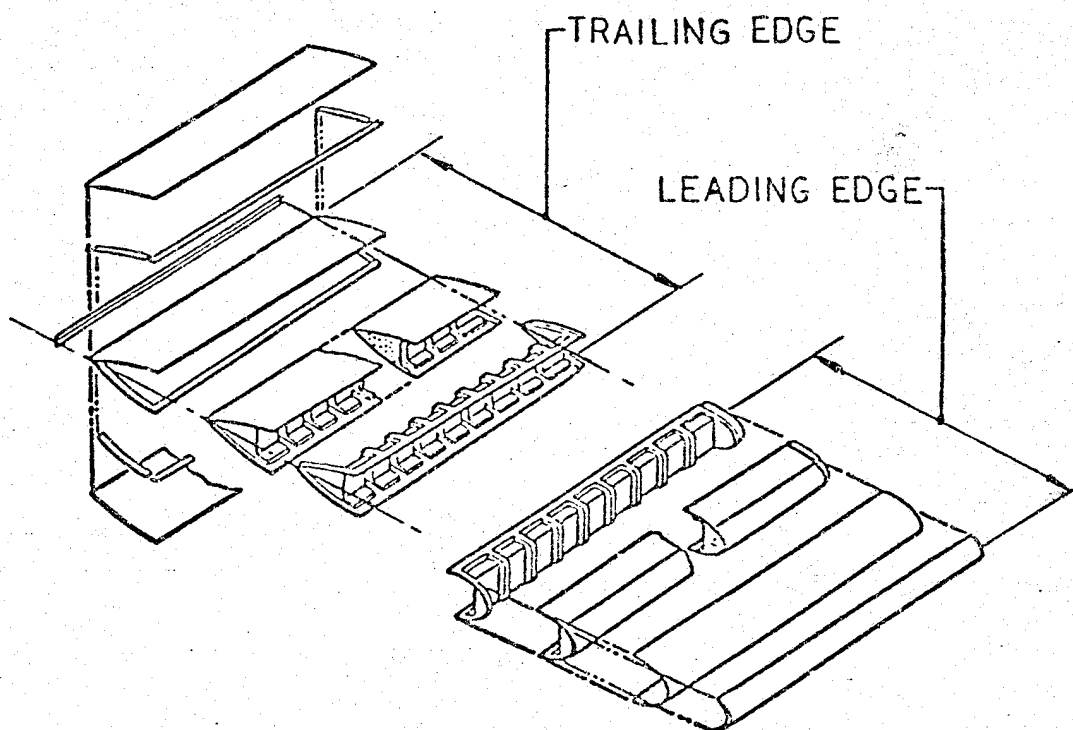
FIGURE 10 - EXPLODED VIEW OF LEADING AND TRAILING EDGE
ASSEMBLIES

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To reduce weight and to improve mass distribution in the blade, the metal spar is cut short and a structural fiberglass tip extension is used. Figure 11 shows a breakout view of the tip parts.
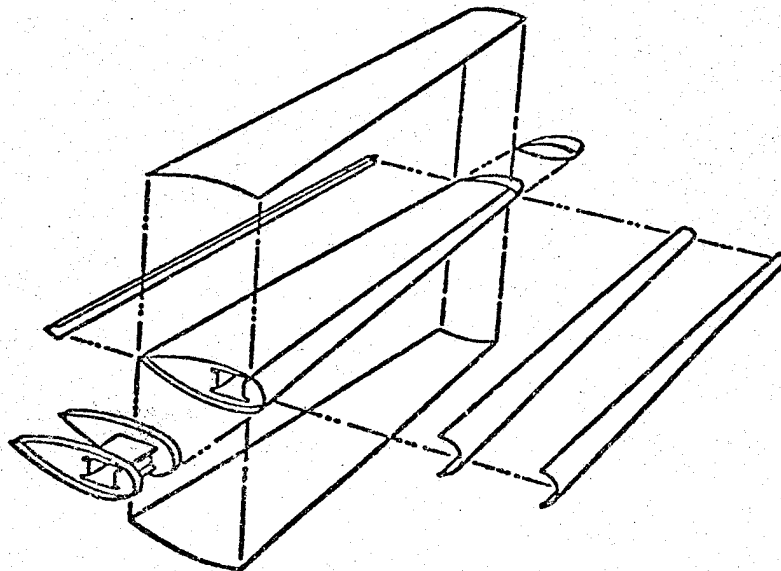
FIGURE 11 - BREAKOUT VIEW OF TIP PARTS

The final assembly of the blade is accomplished in two major steps, as shown in Figure 12. The trailing edge assemblies are bonded to the spar. Progressing from the inboard end of the blade outward, the leading edge sub-assemblies are then attached to the spar trailing edge major assembly by bonding and mechanical fastening. The tip assembly is then bonded and mechanically joined to the blade assembly. The final step is the installation of elastomeric sealing strips between the edges of the fiberglass assemblies to provide aerodynamic sealing of the surfaces.

TRAILING EDGES TO
SPAR ASSEMBLY
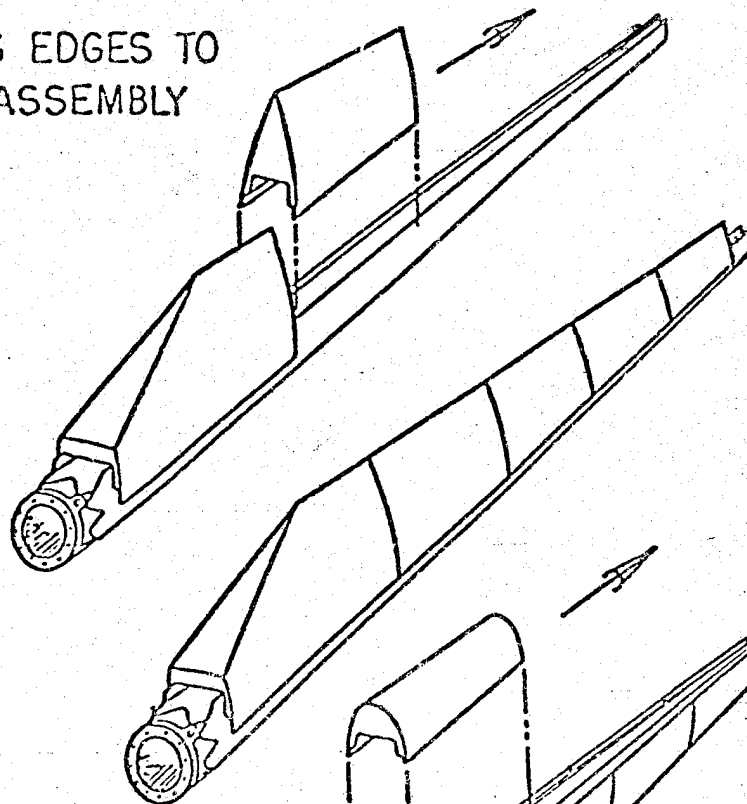
LEADING EDGES AND TIP
TO SPAR ASSEMBLY
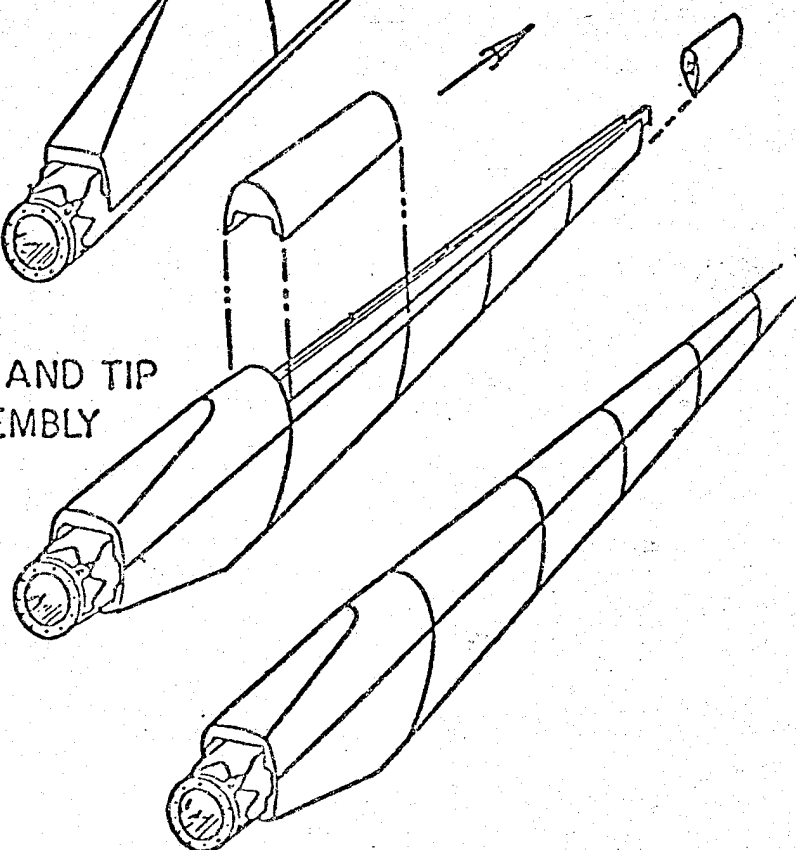
FIGURE 12 - TRAILING EDGES, LEADING EDGES AND TIP TO
SPAR ASSEMBLIES

A study was performed to determine whether the basic concept of the blade design could be utilized through the entire range of blade sizes. The concept of a rectangular spar inside the aerodynamic surface carrying the principal loads with parasitic aerodynamic elements directing the air loads to the spar is applied to the entire range of blades. Figure 13 shows the blades that were included in this study. Using the present configuration with a high aspect ratio blade, the design would use stainless steel for the spar from the 60-foot size blade down to the smaller sizes. The advantage of the stainless steel spar is the ability to taper and spot weld the assemblies together at low cost and with good corrosion resistance for the thinner gage materials needed on the smaller blades. For lengths greater than 60-foot, the design would use high strength, low alloy carbon steel for the spar. The reason for this is that the gage of the materials will be beyond the range of those producible in cold-rolled stainless steels. Thicker gage cryogenic stainless steel material might be used, but we do not think this would be economically feasible in the larger size.

200'

150'

100'   (DIMENSIONS IN FEET)

60'

30'

15'
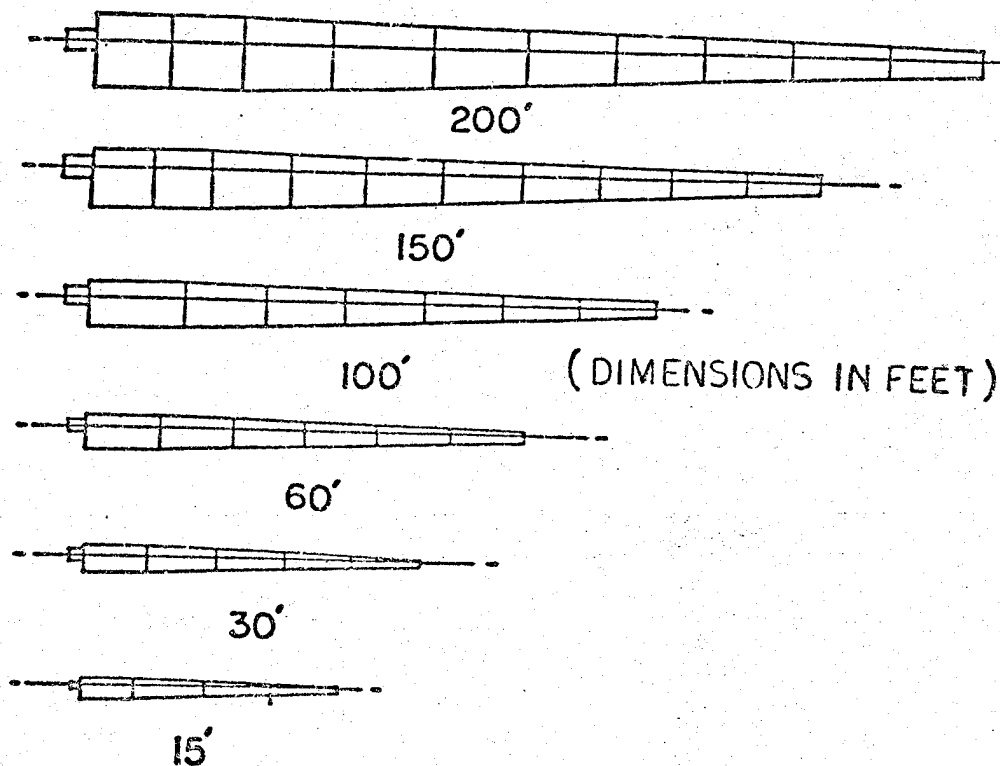
FIGURE 13 - INVESTIGATION OF VARYING LENGTHS

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Figure 14 shows the spar configuration for the 60-foot
blade. This is constructed of 301 1/4 hard .125 thick
stainless steel. Also shown is the configuration of a
spar of 200 ft. length.

SPAR ROOT END OF 60 FT. BLADE

(DIMENSIONS IN INCHES)

SPAR ROOT END OF 200 FT. BLADE
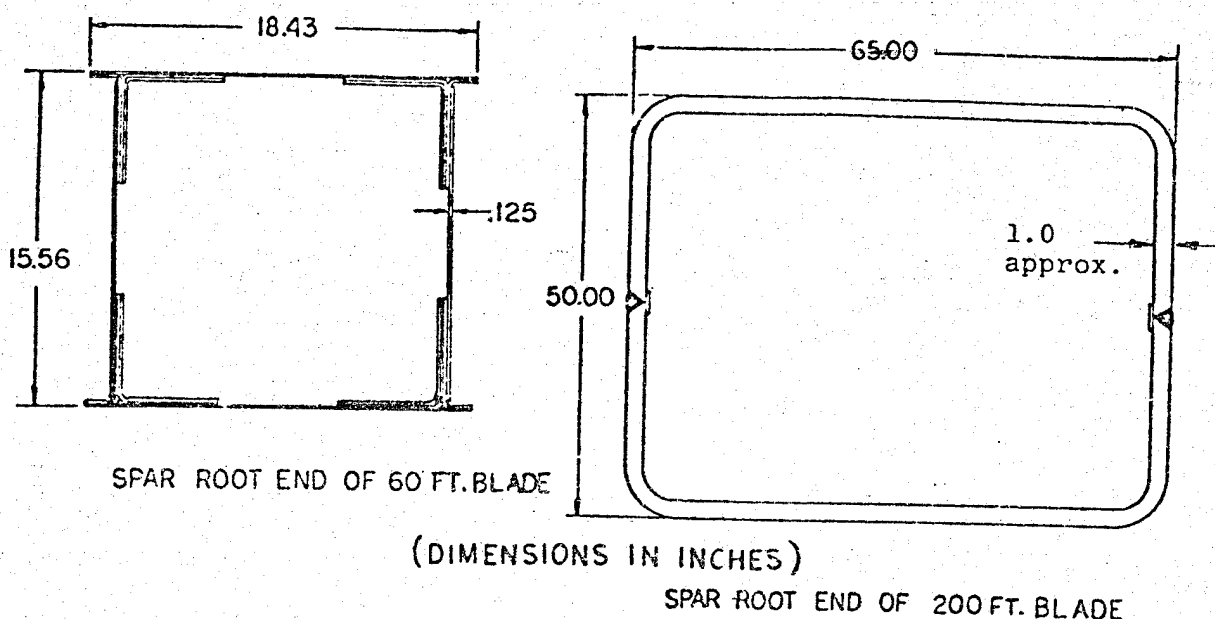
FIGURE 14 - SPAR ROOT END OF 60 FT. AND 200 FT. BLADE

Considering only production costs, it is less costly to
produce a multiplicity of smaller blades rather than a
low quantity of larger blades for the same power output.
Since the blade is only a part of the total system, this
conclusion may not hold when considering the whole system
cost.

Larger blades require manufacturing activities that are
beyond the present state of the art in many areas. To
provide a good, low cost design will require further in-
vestigation. For transportation, a blade up to 85 feet
in length can be shipped in one piece without major diffi-
culty. Greater length blades probably would have to be
shipped in multiple pieces. This, of course, increases
the problems of design of the blade, as this would require
spar joints outboard in the blade that require the same
degree of reliability as the root fitting. This could
add considerably to the weight and cost.

DISCUSSION

The design of large wind turbine blades have conflicting
requirements and criteria. Cost is the most sensitive
requirement and structural reliability is the foremost
criterion. The Budd design is predicated on the premise
that large blades should be an industrial product of pre-
dictable performance and uncomplicated structure. In order
to be successful, rotors must be capable of being produced
in volume at reasonable cost. The Budd Company has exten-
sive experience in the design and manufacture of structures
made of carbon steel, stainless steel, and glass reinforced
plastic. Fabricating techniques combined with a long his-
tory of successful product designs provided the key to
meeting the program objectives.

To achieve a potential 30 year life, the selection and
evaluation of materials used is critical. It became evi-
dent early in the program that design criteria for aircraft
wings and helicopter blades did not apply for a horizontal
axis system. Because of the cyclical loading in the edge-
wise direction, the system is more analogous to a rotating
fatigue machine. With a rotational speed of 40 rpm the spar
structure will experience about 400 million load cycles
during the life of the blade. Existing fatigue data for
most materials are generally limited to the range of 2 to
10 million cycles. It is prohibitively costly and time con-
suming to run fatigue tests to 400 million cycles. Even at
high testing rates it would take many years to obtain actual
test data on full scale test elements; therefore, it was
necessary to develop high cycle endurance strength by a more
practical approach. To accomplish this we chose to establish
the S/N curve, on the basis of test data, in the range of
zero to ten million cycles-to-failure. These S/N curves
were then extrapolated graphically to four hundred million
cycles-to-failure. To determine the fatigue strength for
a different R value (i.e., ratio of minimum to maximum stress)
the modified Goodman Diagram was used. This technique was
applied for both parent material and splices--e.g., spot
welded or butt fusion welded connections. The above des-
cribed procedures have their principle application to base
material properties. When applying this technique to welded
configurations and complete systems, there can be some risk
associated with the selected stress values. Complete sys-
tems have to be tested to provide final proof of the design.
Therefore, a full scale section of the spar, including the
root end, was fabricated and delivered to NASA-Lewis
Research Center for testing.

To supplement the full scale test elements, The Budd Company
using it's own funds, conducted an independent test program
to obtain long term fatigue data for 301 1/4 hard material,

including fusion welding as well as spot welding. The results of this activity were used to provide additional data for the blade program.

The fiberglass leading and trailing edge assemblies that form the aerodynamic surfaces were tested as a complete system, by fabricating a one foot wide section representative of Sta. 187 inches on the blade. The maximum aerodynamic load was simulated by bonding masses to the outer surface of the test element which then was cycled on a shaker test fixture at resonance of the specimen.

Summary data from these test programs are presented in the section on the final design program.

One of the principle outputs required of the contract was to confirm the low cost aspects of the Budd design. To accomplish this, a weight and cost analysis was conducted. The blade final design was processed in detail and manufacturing and tooling costs were determined. This was done for quantities of 2, 10, 100, and 1000 per year. The design was reviewed over the range from 15 ft. to 200 ft. blade size and the blade design proved to be practical. The design study indicated that various spar materials should be used depending on the practicality of application relative to blade size. The detail costing was centered on the Mod. O.A. blade of 60 ft. size. The estimated cost summary for 100 blades per year is shown on Figure 15, and as can be seen in the chart from the NASA specification, the Budd blade falls well within the cost objectives set out by the contract.

The final design presented at NASA-Lewis Research Center, was accepted subject to final test results of the test spar section.

ESTIMATED COST SUMMARY
BASED ON 100 BLADES ANNUALLY

TOOL COST

Tools 1,215,000

Based on Production of 100

Blades Annually

Tool Cost 1,215 Per Blade

(Tools Prorated on)
(10 Years Production)

COST SUMMARY

Material	4,600
Labor	5,570
Tools	1,215
Total	11,385
5% G&A	570
Total Cost	11,955
10% Profit	1,195
Cost & Profit	13,150 Per Blade

$ 13,150.00 PER BLADE
AT *2466 POUNDS
(BASE ON 100 BLADES ANNUALLY)

COST VERSUS WEIGHT DIAGRAM
(COST CURVES FROM BASIC N A S A DATA)

FIGURE 15 - COST SUMMARY

PRELIMINARY DESIGN REVIEW

Task I of the contract consisted of the development of a
preliminary design using the stainless steel/fiberglass-foam
blade concept. Various designs were reviewed, configuration
studies were made, and a preliminary design was completed.
This effort included several design iterations to establish
the applied loads for the blade, which are a function of the
mass distribution of the blade and the blade configuration.
A study of available fatigue data and material properties
was made.

To supplement the Budd Company experience, a subcontract was
released to the Piasecki Aircraft Corp. for review and comment
on the design concept.

Processing and tooling studies were made and a cost study of
the preliminary design was completed. The following data and
reports were submitted as part of the preliminary design
activity:

1. Basic structural and mass distribution of the structure
   for the preliminary blade design

2. The Budd Company's investigation on the fatigue strength
   of spot welded joints in structural stainless steel

3. A study of rigid polyurethane foam

4. Piasecki Aircraft Corp. Report

5. Structural analysis for preliminary blade design

6. Preliminary fatigue properties of materials and connections
   to be used in the Budd Windmill blade

7. Preliminary cost estimate based on 200 units per year

8. Preliminary design review report

9. Preliminary design review presentation

10. Engineering drawing package

The preliminary design review was held at NASA-Lewis Research Center. The proposed blade design was approved. The review panel suggested that the following information be provided to NASA-Lewis prior to the final design review.

a) Analyze in detail all adhesively-bonded joints in the primary and secondary structures.

b) Analyze in detail all spot welded joints between the components of the spar.

c) Analyze in detail the complete structural properties of the spar and fiberglass-foam bodies.

d) Provide a detailed test plan(s) for qualification of all of the spot welded and adhesively-bonded joints.

To the extent possible within the scope of the contract, these items were addressed in the final design.

FINAL DESIGN PROGRAM

The final design program completed a final design of
the blade developed during the preliminary design program
(Task I); the final design program consisted of four
principle tasks.

a) Task II, the first activity of the final design program
contained the fatigue testing and other material evalua-
tions required to verify the design stress levels for
the final design, the design layouts, the structural
analysis, and the design and fabrication of the 20 foot
full scale root end test spar.

b) Tasks III and IV covered the fabrication procedures, cost
analysis, and additional studies to apply the design con-
cept to various size blades.

c) Task V was the preparation and presentation of the final
design to NASA-Lewis Research Center and the preparation
of the final report.

d) The results of the final design program are summarized
in this section of the report. The fabrication of the
spar test element is discussed separately following this
section.

TEST PROGRAM

As mentioned earlier, the achievement of a thirty year life
requires the careful selection and evaluation of materials.
The design operating speed of the Mod-OA wind turbine is
40 rpm, which represents approximately 400 million cycles
of loading during the thity-year life of a blade. It would
be prohibitively costly to perform fatigue tests for 400
milli°n cycles; therefore, it was necessary to use a more
pra   l approach, similar to what other people have assumed.
We a   e that the fatigue strength at ten million cycles is
equi _nt to the endurance limit. In other words, we assume
the S/N curve is flat by ten million cycles. Also, since
few tests were performed, we took the lower limit of the test
data as a measure of "test value" and 80% of that value as
the "design allowable strength".

Considerable difficulty was encountered in developing a
design for the end connections for the full scale test elements.
Many early specimens failed prematurely at the pin connection
of the specimen to the test machine.

The high stress levels achieved on the test configurations
contributed to the problem. The solution to the end connec-
tion problem incorporated a split bushing arrangement with
through bolt to provide a friction bearing clamp to carry
the test loads. The design developed is shown in Figure 16.

CLAMP BUSHINGS
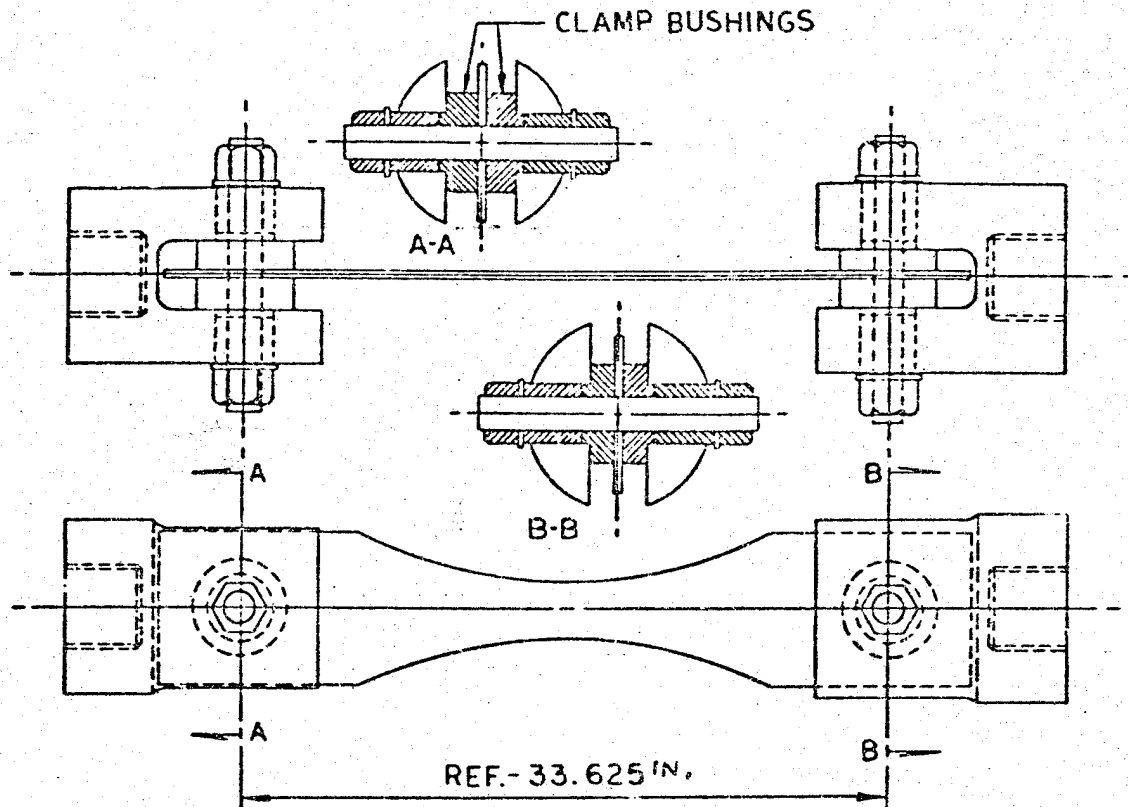
A-A

B-B

REF.- 33.625 IN.

FIGURE 16 - TYPICAL FATIGUE SPECIMEN CONFIGURATION

The logic applied to the test program required that we con-
duct satisfactory fatigue tests for the base material and
each joint configuration used in the design and that this
data would be used to provide final sizing of the spar
structural elements.

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Some difficulties were encountered on two of the preliminary
design joints. The base line design had a bolted and bonded
root end attachment of the spar to root fitting. The use of
mechanical fasteners was not a successful design, due to the
stress concentrations generated by the first bolt in the
load system. The design was changed to a welded connection
that was satisfactorily fatigue tested. The other area of
the base line design that had to be changed was the method
of stopping off the reinforcement angles in the corners of
the spar. It was found by testing that it was not possible
to eliminate the stress concentration on the last spot weld
at the end of the reinforcement. Therefore, the angles were
tapered and extended the full length of the spar to eliminate
the stress concentration. To improve the mass distribtuion
in the spar, the reinforcement angles were butt fusion welded
in two places allowing a reduction in the gage thickness out-
board on the spar.

To supplement the full scale test elements, The Budd Company,
using its own funds, conducted an independent test program
to obtain long term fatigue data for the 301 1/4 hard material,
including fusion welding as well as spot welding. The results
of this activity were used to provide additional data for the
blade design. Previous Budd Co. data was limited to approxi-
mately 2 million fatigue cycles. It was necessary to extend
this data to 10 million plus load cycles to determine if the
S/N curve was essentially flat beyond 10 million cycles.
This was accomplished to our satisfaction, and the data was
adjusted by use of modified Goodman diagrams to the proper
R value and applied to the spar design. The data marked by
an * on Figure 17 was obtained from the Budd supplemental
program.

The data from the fatigue testing indicates that the knee of
the S/N curve occurs at less than 4 million cycles for both
the material and joints used in the design. This data supports
our premise that 10 million cycles of fatigue testing is
sufficient to predict long-term durability for the product.

Some of the tests were run for more than ten million cycles.
The design allowable strength was defined as 80% of the test
strength to allow for possible scatter. Also, the design
allowables for other R values was established by using the
Goodman Diagram.

There are five principle joint designs in the spar. A summary
of the test results are shown in Figure 17. The data from
the fatigue test program was presented at the final design
review meeting and was included in the final design review
report submitted to NASA.

25

TEST PROGRAM CONCLUSIONS
OF 301 1/4 HARD STAINLESS STEEL AND 4130 STEEL
SUMMARY OF TEST LEVELS AT 10 MILLION OR MORE LOAD CYCLES
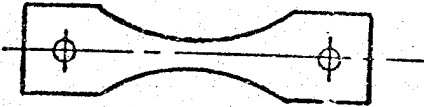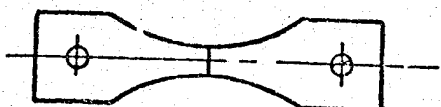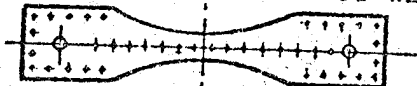### ADJUSTED FOR "R" VALUE OF -.5
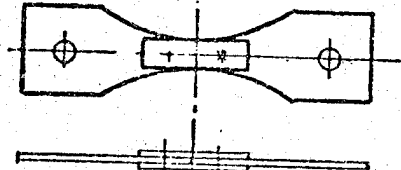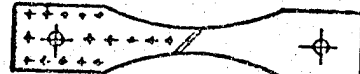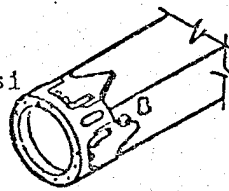
TEST	ADJUSTED TEST VALUE	DESIGN ALLOWABLE (80% of ADJUSTED TEST VALUE)
	SHOWN IN CYCLIC STRESS	

BASE METAL (301 1/4 H STAINLESS STEEL)

     ✳ ± 49,000 psi   ± 39,200 psi

BASE METAL (BUTT ARC WELDED & COLD WORKED)

     ✳ ± 24,000 psi   ± 19,200 psi

BASE LINE SPARE -SPOT WELDED (3 LAYERS)

     ± 21,000 psi   ± 16,800 psi

MAX SHEAR LOADS IN SPOT WELDS (R=+.1)

     ✳ + 562.5 (CYCLIC LOAD PER WELD) SINGLE SHEAR   + 450 (CYCLIC LOAD PER WELD) SINGLE SHEAR

ROOT END ATTACHMENT - ARC WELDED

     ±16,000 psi   ± 12,800 psi

A-TEST LOAD DETERMINED BY STRESS LEVELS AT AREA "A"
B-AREA REPRESENTED BY TEST SPECIMEN

4130 CHROME MOLY STEEL   + 38,000 psi (CHART VALUE)   ± 30,400 psi

FIGURE 17 - TEST PROGRAM CONCLUSIONS

The fiberglass leading and trailing edge assemblies that
form the aerodynamic surfaces were tested as a complete
system by fabricating a one foot wide section representative
of Sta. 187 inches on the blade. The maximum aerodynamic
load was simulated by bonding masses to the outer surface
of the test element which was then cycled on a shaker test
fixture at resonance of the specimen. Figure 18 shows the
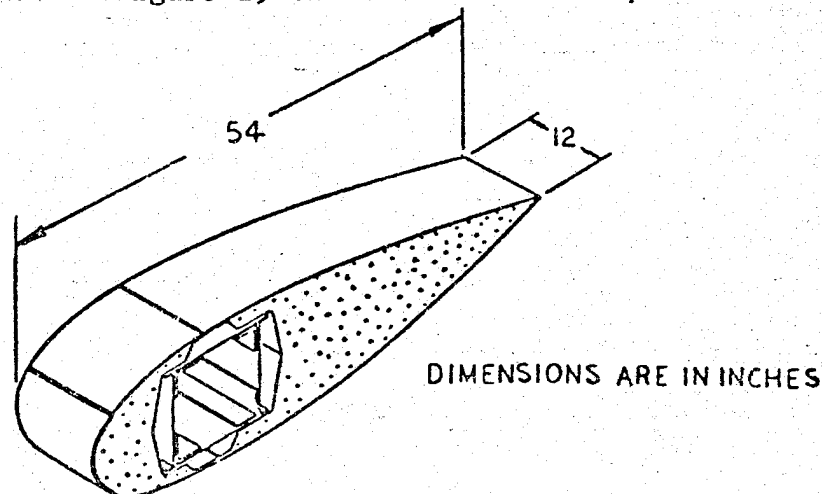test specimen and Figure 19 shows the test setup.

DIMENSIONS ARE IN INCHES

FIGURE 18 - TEST SPECIMEN AERODYNAMIC SURFACE
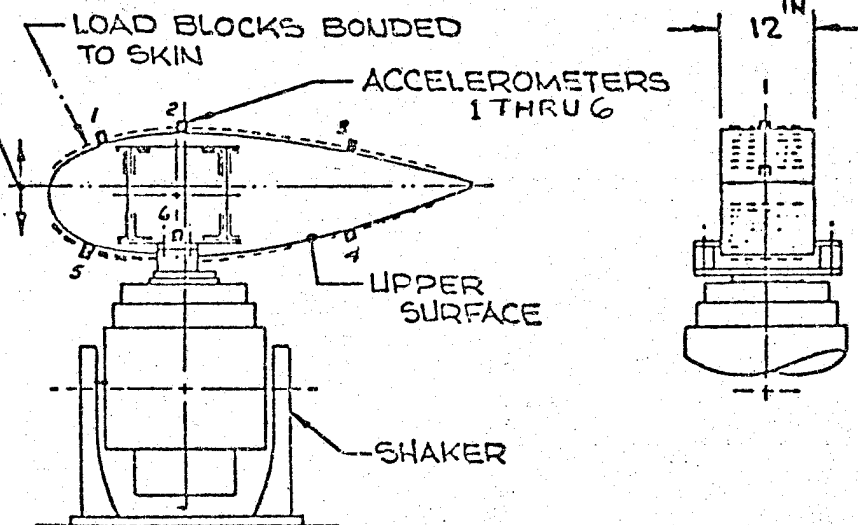
FIGURE 19 - AERODYNAMIC LOADING OF BONDING SURFACES

It was required that the blade sustain an ultimate load of
50 pounds/sq.ft. on the aerodynamic surfaces. To demonstrate
this capability, a static test was performed. Figure 20
shows the test method. Sand bags were used to load the
specimen.

NOTE: ① THRU ④ ARE DIAL GAGE LOCATIONS
W X Y Z ARE STRAIN GAGE LOCATIONS
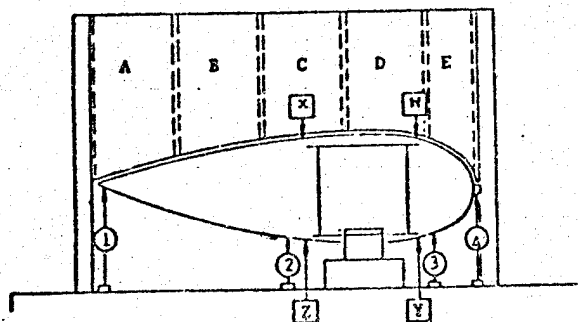
FIGURE 20 - STATIC LOAD TEST (CONFIGURATION)

The aerodynamic surfaces are constructed from the following
materials:

Basic Material .... Glass Reinforced Polyester
                               with 17% minimum glass
                               content

Trailing Edge Skins .... .030 inches Commercial Sheet
                               (as above)

Leading Edge Skins .... Commercial Lay-up
and Inner Channels   (.060 to .125 inches)

Foam (2-1/2 lb. Density) .... Rigid Urethane Foam
                               (Mobay Chemical Co.
                               NB-2379 36A)

Adhesive .... 3-M Two Component Adhesive
                               (EC-3549 BA)

The following is a summary of the results of the test
program for the aerodynamic surfaces.

FATIGUE TEST

Test #1

Aerodynamic load equivalent from 0 to design max.
Mass load equivalent from 0 to 2 G
Tested at resonance (approx. 18 cycles per second)
Tested to 11,000,000 cycles (no failure)

Test #2

Aerodynamic load was increased to -1 to +2 x design max.
Mass load was increased to -2 to +4 G
Tested at resonance (approx. 18 cycles per second)
Tested to 1,100,000 cycles
(Induced minor failure in urethane bond to spar)

STATIC TEST

Static Design Load 50 lbs. per sq. ft. Proof Load

Conducted without repair

Test 1 - Loaded to 175 lbs. per sq. ft. (no failure)

Test 2 - 24 hr. Creep Test at 175 lbs. sq. ft.
(no creep)

The test program indicates that the design for the fiber
glass aerodynamic surfaces is more than adequate for the
design. Some weight reduction could be made although this
is difficult due to process limitations.

Supplemental tests were run to evaluate the static ultimate strength of the bond between the foam core material and the skin (see Figure 21) and the ultimate strength of the aerodynamic surface bond to the spar and its elastomeric characters. See Figure 22.

FIBERGLASS-POLYESTER SKIN

ATTACH FITTING

2.5 PCF FOAM

12 SQ. IN.

SKIN TO FOAM CORE (1ST FAILURE)
PULL STRENGTH 35 LB/SQ INCH
TEST CONTINUED TO SEPARATION OF SKIN & ATTACH FITTING (URETHANE BOND)
PULL STRENGTH 137 LB/SQ INCH

FIGURE 21 - SKIN TO FOAM PULL STRENGTH TEST

FIBERGLASS ANGLE (OUTER ASSEMBLY)
URETHANE ADHESIVE
S/S SHEET (SPAR)
.20 IN.
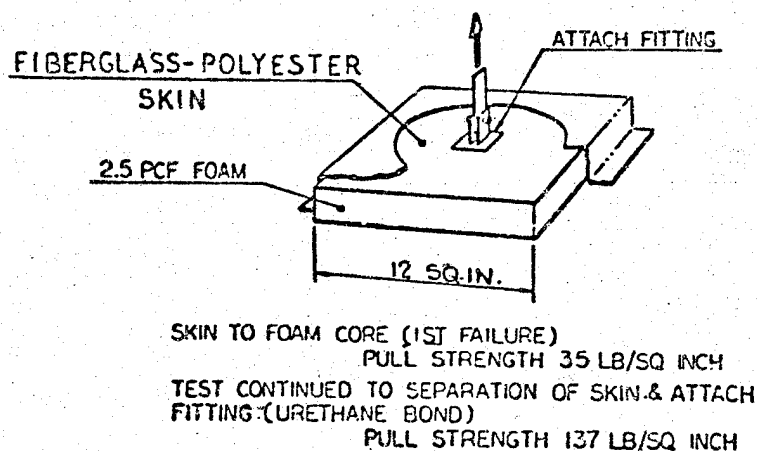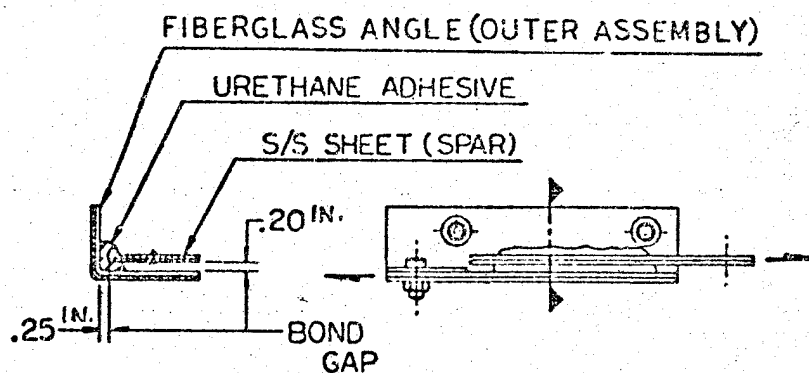.25 IN.
BOND GAP

BOND TO STAINLESS STEEL ~1,340 LB./SQ. IN. (NO FAILURE)
BOND TO FIBERGLASS ~ 847 LB./SQ. IN. (FAILURE)
DEFLECTION AT FAILURE ~ 0.3 INCHES

FIGURE 22 - ULTIMATE LOAD SHEAR TEST (BOND OF THE SPAR TO AERODYNAMIC SURFACES)

Data for the design allowable to be used for the skin
(.060 inches polyester fiber glass cloth sheeting) for
the tip extension is taken from the Military Handbook
MIL-HDBK-17 date November 5, 1959, Plastics for Flight
Vehicles, Part I - Reinforced Plastics. Based on an
ultimate strength of 30,000 psi per square inch from
the handbook and a maximum design calculated mean stress
of 3,600 psi and a cycle design stress of 1,900 psi the
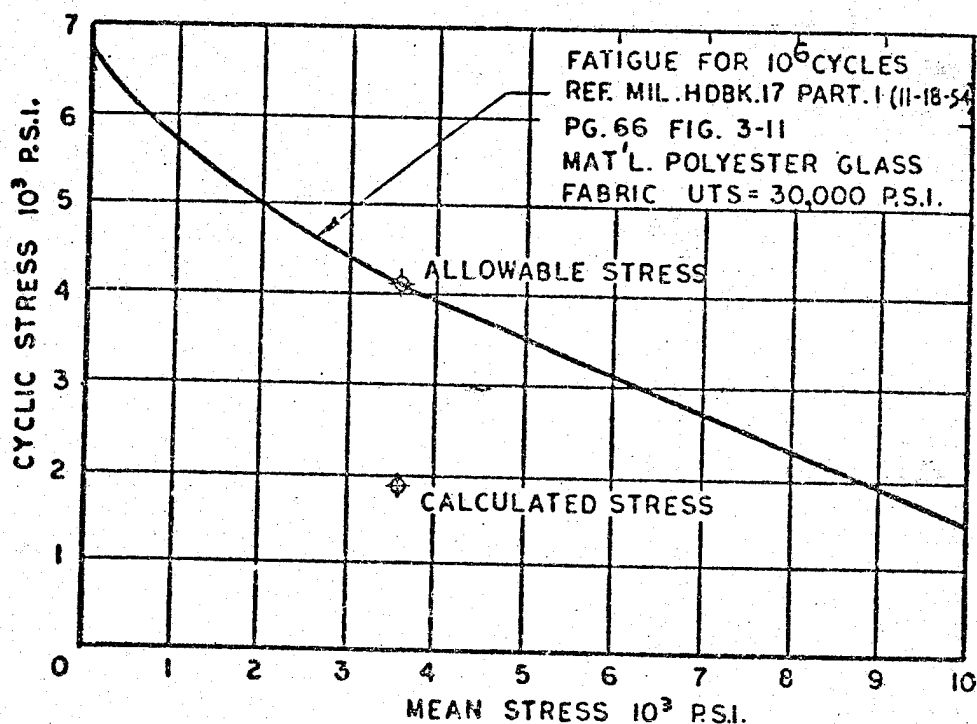allowable stresses shown in Figure 23 can be determined.

FIGURE 23 - FATIGUE FOR 10^6 CYCLES (POLYESTER GLASS)

In conclusion, we believe that we have accomplished a
highly comprehensive test program, considering the limited
funding available. Many developmental tests were conducted
in all areas to provide the summary data.

STRUCTURAL ANALYSIS

Upon completion of the test program, the preliminary design
was modified to be consistant with the test data. This
included the change to a welded attachment of the root
fitting and modification of the spar reinforcement angles
to run the full length of the spar. The first iteration
of the mass distribution was run on the MOSTAB computer
code by NASA-Lewis to establish revised loads. Stress levels
were developed from these loads and a second design iteration
was made which reduced the mass of the spar and moved the
center of gravity closer to the root end.

The feasibility of these modifications was demonstrated by
the results of the 301 test program (which is discussed in
Section on Test Program) - particularly the fatigue strength
of butt fusion weld splices, which permits the reduction of
the gage of the spar plates from .125 to .093 to .060 inches
in three steps proceeding on board.

The mass distribution on this design was run on MOSTAB at
NASA-Lewis and resulted in reduced loads in both axes of the
spar. At this time, although the design met the contract
requirements, NASA-Lewis reported that the design could not
be run on the Mod-OA system unless the root end mass
bending moment could be reduced to 47,000 ft. lbs. due to
limitations established from experience gained running these
machines.

The design was modified to meet the new NASA guidelines by
shortening the stainless spar and providing a light-weight
fiberglass tip extension from Sta. 650 to the outboard end
of the blade. This modified mass distribution resulted in
a mass moment for the final design of under 46,000 ft. lbs.
The mass distribution for this design was run on MOSTAB by
NASA-Lewis. Final loads were established and the analysis
of the design was conducted using that data. Figure 24 shows
the mass distribution on the blade.

Figures 25 and 26 show the edgewise and flatwise
bending moments supplied as the output of the NASA MOSTAB
run. These curves specify the steady and cyclic moments
and therefore represent the fatiguing loads for the ana-
lysis and design of the spar. The blade planform and sec-
tion along the span determined the chordwise spar location
for the best combination of bending and polar moments of
inertia.

The foam core and fiberglass skin which comprise
the leading and trailing edge structures were designed
and analyzed using data from the NASA curves showing
the chordwise load distribution. The magnitude of the
loads on the leading and trailing edges were determined
from the above flatwise bending moment curves by measur-
ing the slope of such curves at any given station. Thus,
the slope at stations bracketing a desired location gave
the difference in shears, which is the load within this
portion of the span. The load, for twelve inches of
span, was then distributed chordwise on the design.
A full size test model of the twelve-inch width section
was made for Sta. 187 and successfully survived the
fatigue tests which simulated actual loading of the blade.

The stainless steel spar was located with its cen-
troidal axis around the 30% chord in order that the maximum
depth shall be realized. During stages of the design
effort, fatigue allowables for stainless steel sheet and
spotwelds were used for sizing the spar sections so they
may endure the flatwise and edgewise bending moments.
Goodman Diagrams were constructed for representing these
fatigue allowables. The most critical portion of a typical
section of the spar appeared near the extreme corners where
spotwelds attached the upper and lower plates to the verti-
cal webs. This is shown schematically in page 43 as the
"spar final connection assembly".

At any given station, the steady and cyclic bending
moments were known about each principal axis and therefore
the mean and cyclic stresses at any point within the spar
section became the algebraic sum of like stresses (steady
or cyclic) produced by each bending moment. The system
of reinforcing members was designed to assure that each
spar section would have sufficient properties to keep
stresses within reasonable values for minimum weight. As
the design progressed, Budd testing of stainless steel
specimens and spotwelds provided more current fatigue in-
formation which was immediately employed so that stresses
remained within the newly-constructed Goodman Diagram.

Initially, the stainless steel spar extended to the outer tip with a minimal section at very low stresses from Sta. 650 outboard. This suggested the use of a complete fiberglass foam assembly without the stainless steel spar. Analysis indicated that the fiberglass skin provided enough section properties to resist fatigue stresses caused by the outboard bending moments. The flatwise mode was most critical for stresses around the 30% chord at Sta. 650 and were maintained at levels far below fatigue allowables for fiberglass (see Figures 2 and 28). Edgewise moments produced stresses at the leading and trailing edges of low magnitude due to large section properties in the edgewise direction. In view of the conservative material allowables used for this portion of the blade. a great weight saving was still achieved through judicious design and analysis.
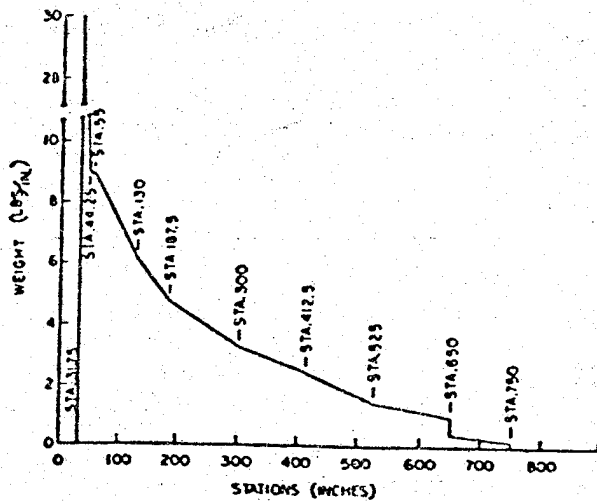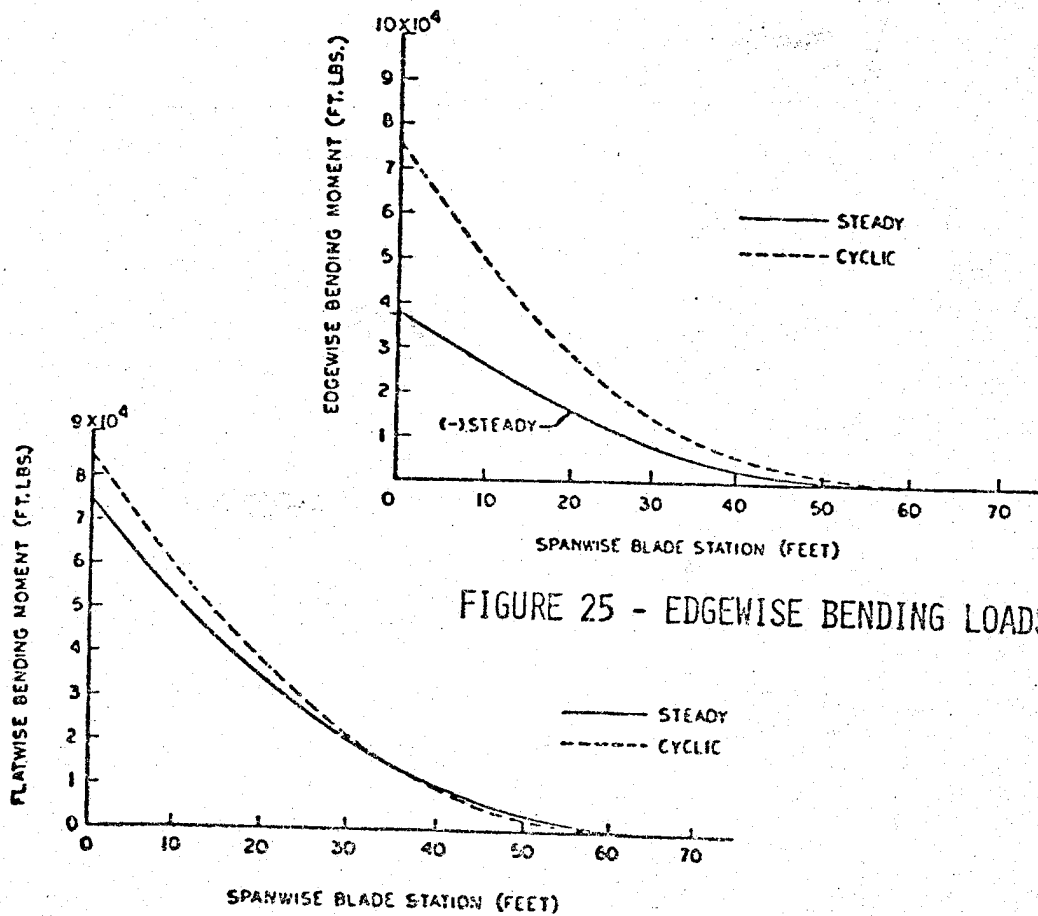
FIGURE 24 - BLADE MASS DISTRIBUTION

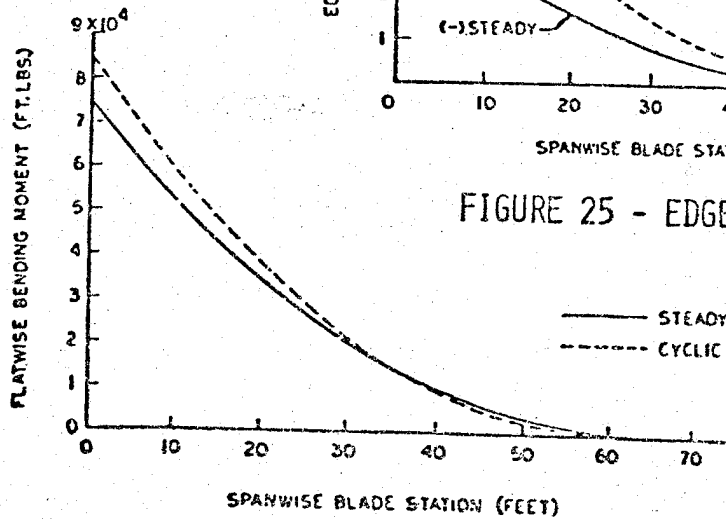FIGURE 25 - EDGEWISE BENDING LOADS

FIGURE 26 - FLATWISE BENDING LOADS

The spanwise moments of inertia of the spar were calculated
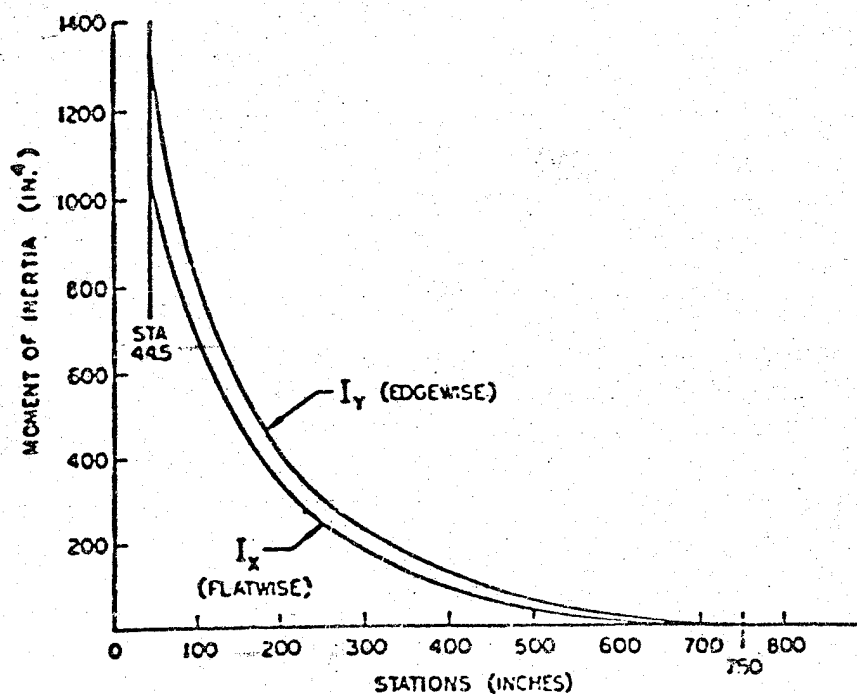and are shown on Figure 27.

FIGURE 27 - SPAR MOMENT OF INERTIA

The chart shown on Table 1 is a summation of the calculated
stresses developed in the spar.
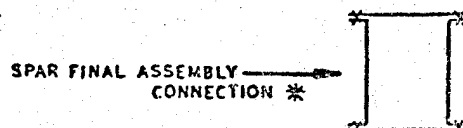
STA.	WEB	CAP	ALLOWABLE CYCLIC - STRESS(PSI)	MAX.COMBINED CYCLIC - STRESS(PSI)	MEAN STRESS(PSI)	MAXIMUM STRESS(PSI)	MINIMUM STRESS(PSI)	$R = \frac{MIN.}{MAX.}$	SHEAR ✳ FLOW (LB/IN)	MARGIN OF SAFETY
445	.125	.125	12,800	11,000	3,340	14,340	-7,660	-.53	103±13	164 %
187.5	.125	.125	16,800	15,914	4,814	20,728	-11,100	-.54	102±15	5.6 %
300	.090	.125	16,800	16,000	5,285	21,285	-10,715	-.50	104±32	5.0 %
412.5	.090	.090	16,900	14,300	6,753	21,053	-7,547	-.36	85±32	17.5 %
525	.060	.060	15,800	12,383	7,790	20,173	-4,593	-.23	73±35	35.7 %
637.5	.060	.060	16,800	6,959	6,278	13,237	-621	.05	74±05	141.4 %

TABLE 1 - SUMMATION OF STRESSES IN SPAR

The structural characteristics of the blade were run on a
computer program at Budd and at NASA-Lewis and these pre-
dicted a flatwise natural frequency 1.86 hz and a edgewise
natural frequency 2.15 hz. NASA-Lewis approved these values
at the final design review.

Figure 28 shows a section of the tip extension at the area of
highest bending stress.

.080 INCH
FIBERGLASS
LAYUP

.060 INCH
GLASS CLOTH SHEET

EPOXY BOND

ELASTOMERIC
ABRASION SHIELD

I_y 211 in.^{4}

I_x 13.7 in.^{4}

MAXIMUM
COMBINED STRESS
3,600 P.S.I. MEAN
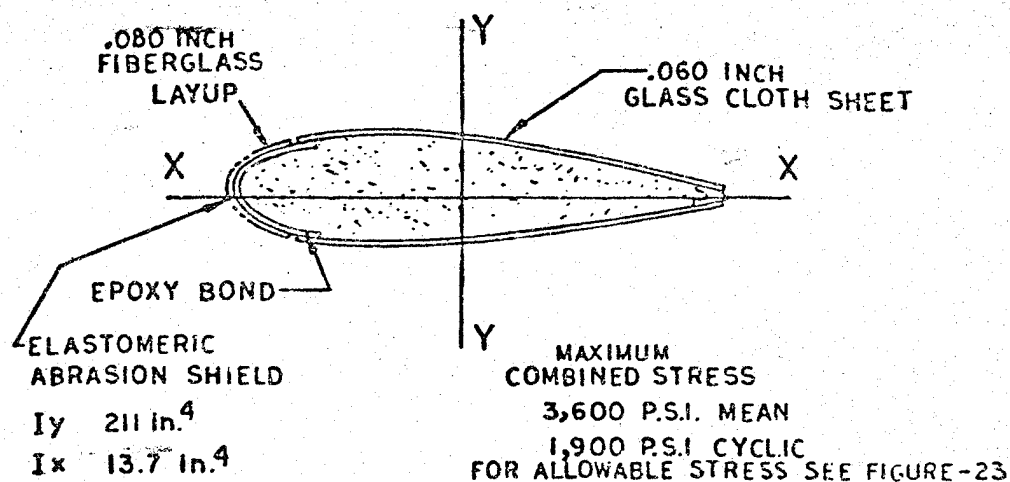1,900 P.S.I CYCLIC
FOR ALLOWABLE STRESS SEE FIGURE-23

FIGURE 28 - SECTION FIBERGLASS OF TIP EXTENSION

The complete structural analysis was submitted to NASA-
Lewis at the final design review. The full scale root end
test beam was designed and the analysis was submitted during
the final design review. The test beam will be discussed in
more detail in the section on fabrication of the test beam.

COST AND WEIGHT ANALYSIS

We have made a cost and weight analysis of the blade design.
The blade assemblies and parts were reviewed for manufactur-
ing procedures and tooling and weights were determined.
Figure 29 shows the basic configuration for the 62 ft. blade.
A summary of the weight and cost of each major assembly,
including tooling cost, is presented in Table 2. The costs
are based on a production level of 100 blades per year.
Figure 32 shows the cost summary of the blade design at 100
units per year and the relative cost of the blade versus the
requirements established by the contract.

As can be seen, we fall well within the low cost objectives
established by NASA-Lewis. Table 3 shows costs established
to predict the approximate blade cost at various production
levels per year. These values have been rounded off to
simplify the numbers. The process work done and the tooling
studies indicate the design is a practical low cost approach
to wind turbine blade design.

SECTION-CC

NACA 23016

SECTION-BB

NACA 23030

SECTION-DD

ROOT FITTING

NASA ROOT FITTING (REF.)

.688 D. - 24 HOLES
1.125 D. SPOT FACE

SECTION-AA

20" O.D. - 13" I.D.

15.96 DIA.
15.97

18.62 B.C.

DIMENSIONS ARE IN INCHES.

PLAN VIEW OF BLADE

90% LINE
& & SPAR
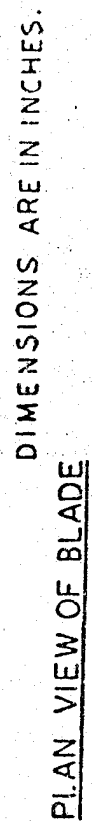
FIGURE 29 - 62 FT. BLADE CONFIGURATION

16.71 SPAR
16.84 ROOT FITTING

(DIMENSIONS ARE IN INCHES)

— SPOTWELDS

8.50

PRECISION GRIND AND
REMOVE ALL UNDER —
CUTTING AT WELD TOE.

SHOT PEEN - BOTH SIDES
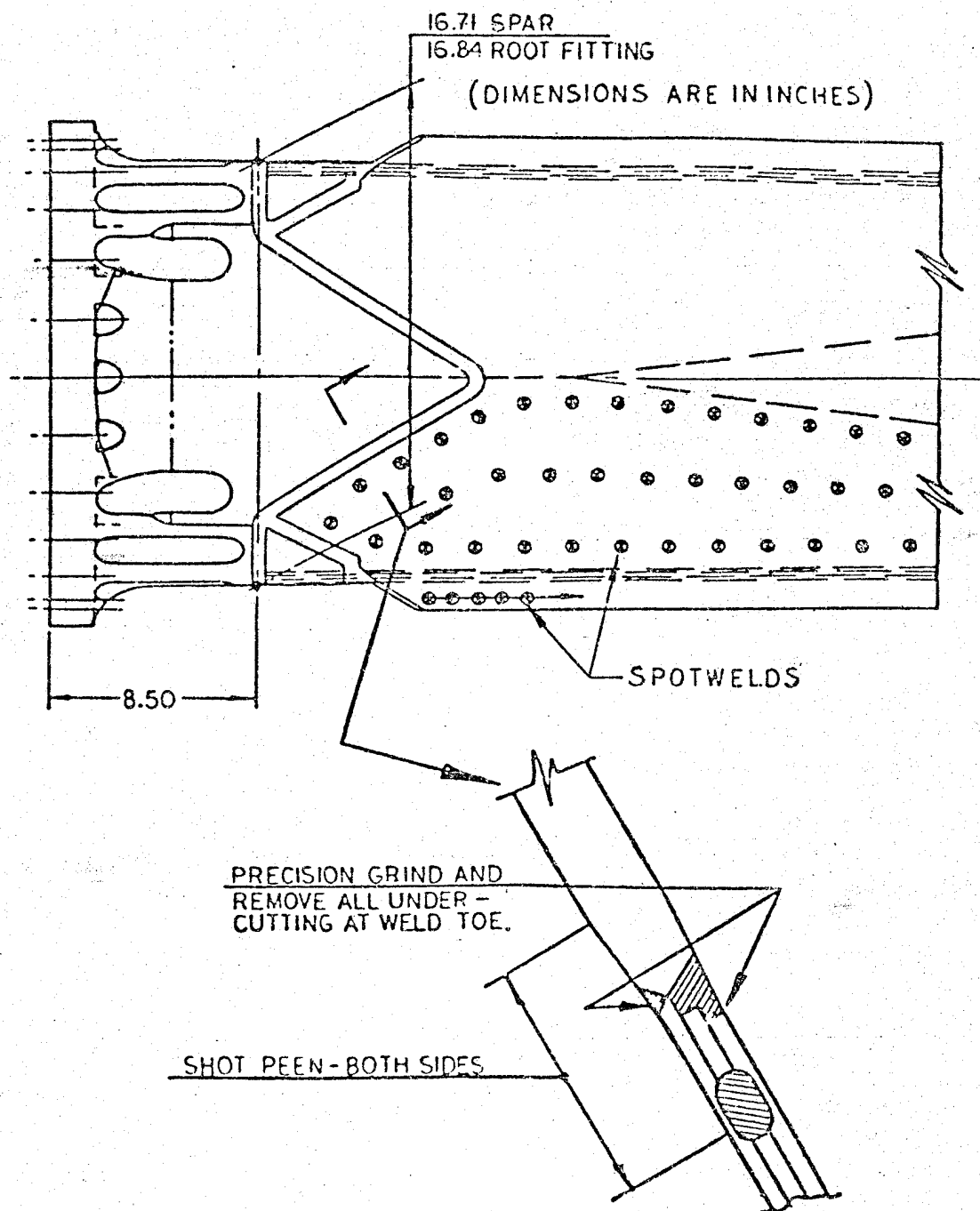
FIGURE 30 - ROOT END - SPAR CONFIGURATION

40

ELASTOMERIC BOND
4 CORNERS
3 INCHES LONG ON
APPROX. 12 INCH SPACING

2.17°

16.20 IN.

54.00 IN.
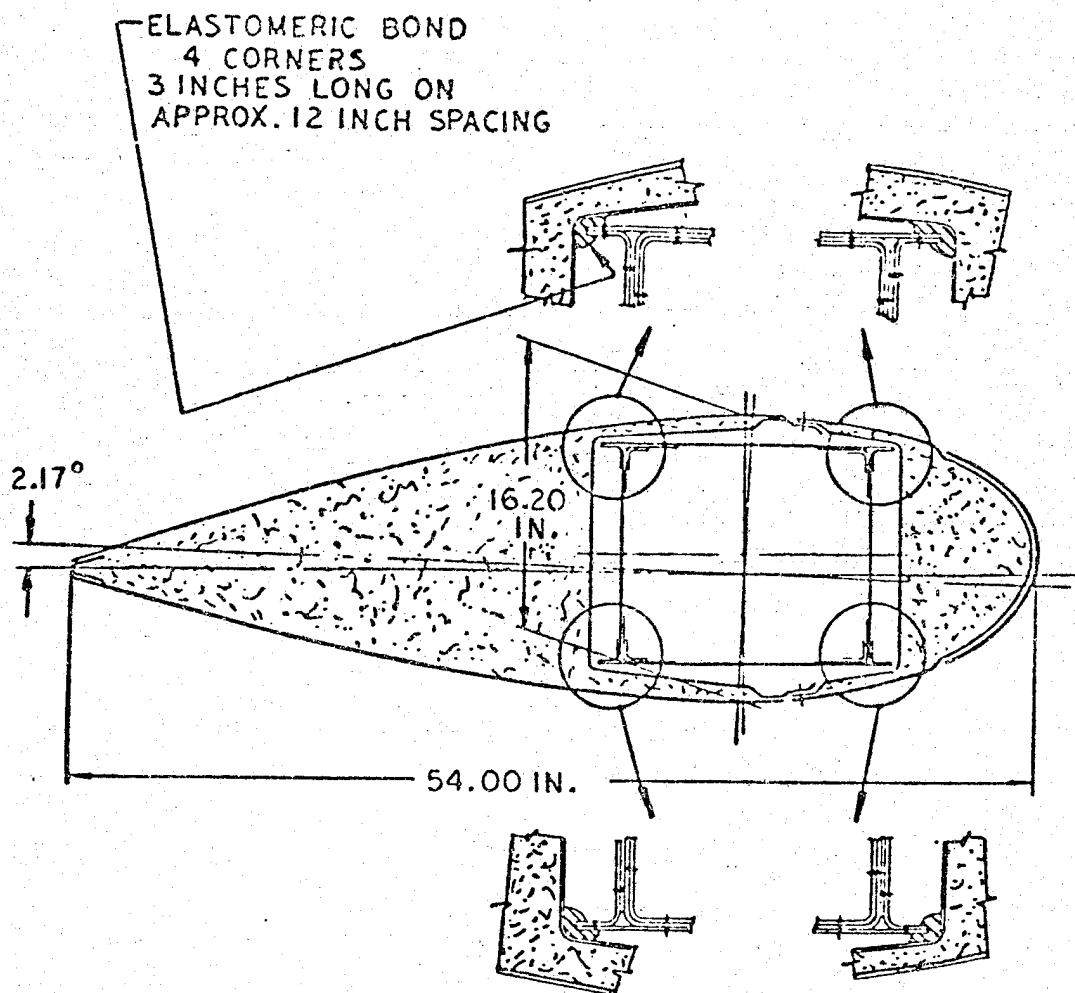
FIGURE 31 — SECTION THRU BLADE — OUTBOARD OF STATION 187.5

PART.NO.	DESCRIPTION	QTY	WEIGHT	MAT'L	LABOR	COST	TOOL COST
0518-100101	Spar Assembly	1	1635.7	2442	2904	5346	441,300
104	Trailing Edge Assy	1	86.4	222	176	398	54,500
105	" " "	1	66.6	173	172	345	50,800
106	" " "	1	55.1	135	150	285	45,500
107	" " "	1	45.3	114	141	255	40,300
108	" " "	1	33.4	84	128	212	36,000
121	Leading Edge Assy	1	73.6	287	86	373	41,000
122	" " "	1	52.7	253	86	339	40,000
123	" " "	1	40.9	170	75	245	34,000
124	" " "	1	30.9	138	66	204	30,500
125	" " "	1	21.5	107	60	167	27,800
126	Fiberglass Tip Assy	1	43.8	110	146	256	45,400
145	Joint Seal	5	12	10	-	10	-
146	Holding Brkt Boot	2	2	4	-	4	1,000
147	Upper Root Fitting Cover	1	7	24	5	53	5,200
148	Lower Root Fitting Cover	1	7	24	5	53	5,200
150	Spar Root Fitting Assy	1	1815.7	15	.275	290	36,000
200	Root Fitting	1	180	169	396	565	66,000
	Adhesive	A/R	72.1	66	-	66	-
100	Blade Assy	1	-	53	704	757	219,700
TOTAL			2466	4600	5570	10170	1,215,000

TABLE 2 - COST ANALYSIS SUMMARY (BASED ON 100 BLADES
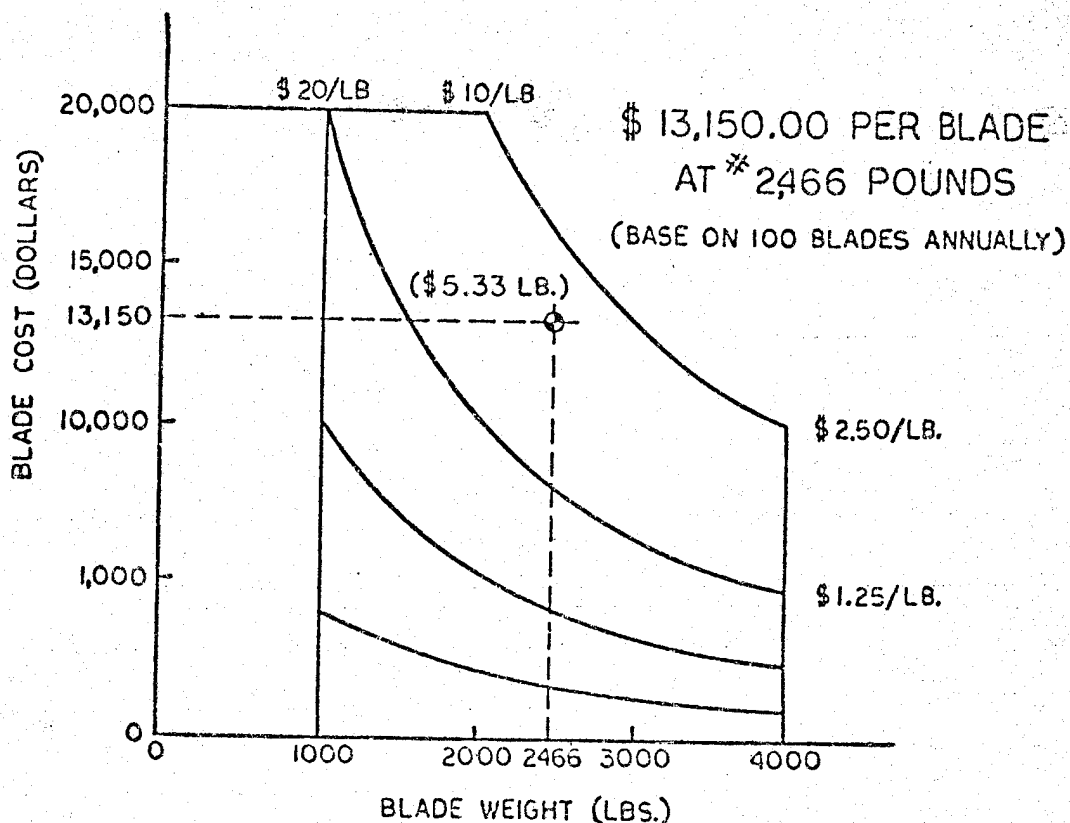ANNUALLY)

42

ESTIMATED COST SUMMARY
BASED ON 100 BLADES ANNUALLY

TOOL COST

Tools 1,215,000

Based on Production of 100

Blades Annually

Tool Cost   1,215 Per Blade

(Tools Prorated on
10 Years Production)

COST SUMMARY

Material	4,600
Labor	5,570
Tools	1,215
Total	11,385
5% G&A	570
Total Cost	11,955
10% Profit	1,195
Cost & Profit	13,150 Per Blade

COST VERSUS WEIGHT DIAGRAM
(COST CURVES FROM BASIC N A S A DATA)

FIGURE 32 - COST SUMMARY

43

MANUFACTURING COST FOR VARIOUS BLADE QUANTITIES

COST CURVED FOR VARIOUS BLADE QUANTITIES
BASED ON 1978 DOLLARS

QUANTITY	COST PER BLADE (EXCLUDING TOOL COSTS)
2	$ 60,000
10	$ 40,000
100	$ 11,500
1000	$ 8,800

TOOLING COST BASED ON PRODUCTION TOOLS
FOR ALL QUANTITIES

PRODUCTION TOOLS	1,215,000
5% G & A	60,750
TOTAL COSTS	1,275,750
10% PROFIT	127,575
COSTS & PROFIT	1,403,325

TOOL COST TO BE AMORTIZED BASED
ON THE QUANTITY OF BLADES TO BE BUILT

NOTE: UP TO 20 BLADES THERE IS A COST ADVANTAGE TO
USE PROTOTYPE TOOLS VS PRODUCTION TOOLS

TABLE 3 - MANUFACTURING COST OF VARIOUS BLADE QUANTITIES

SHIPPING AND HANDLING

A study was made to determine a practical means for handling and shipping the blades. Figure 33 shows the shipping fixture as visualized. We would consider shipping the blades either by truck or by rail. There would be a mounting fixture for the root end of the blade and a mounting fixture for the midpoint location of the blade. These fixtures are designed so that any deflection of either the trailer or the railcar flat bed would not put extraneous loads onto the blade during shipment. When shipping, the blades would be wrapped in polypropylene plastic material to protect them from the weather and vandalism. On railcar shipment, it would be desirable to have a gondola type car with high sides for protection. Figures 34 and 35 show the lifting provisions made at the root end and at the mid-spar point. The blade can be lifted vertically with the leading edge or trailing edge up and it can be lifted with the blade in the horizontal position. There are four lifting lugs on the root end fitting. The root end fitting at this location has been increased in thickness and the lugs are in a low stress area. Double lifting lugs are provided at the mid-blade point to provide capability of transfer from one lift system to a second lift system.
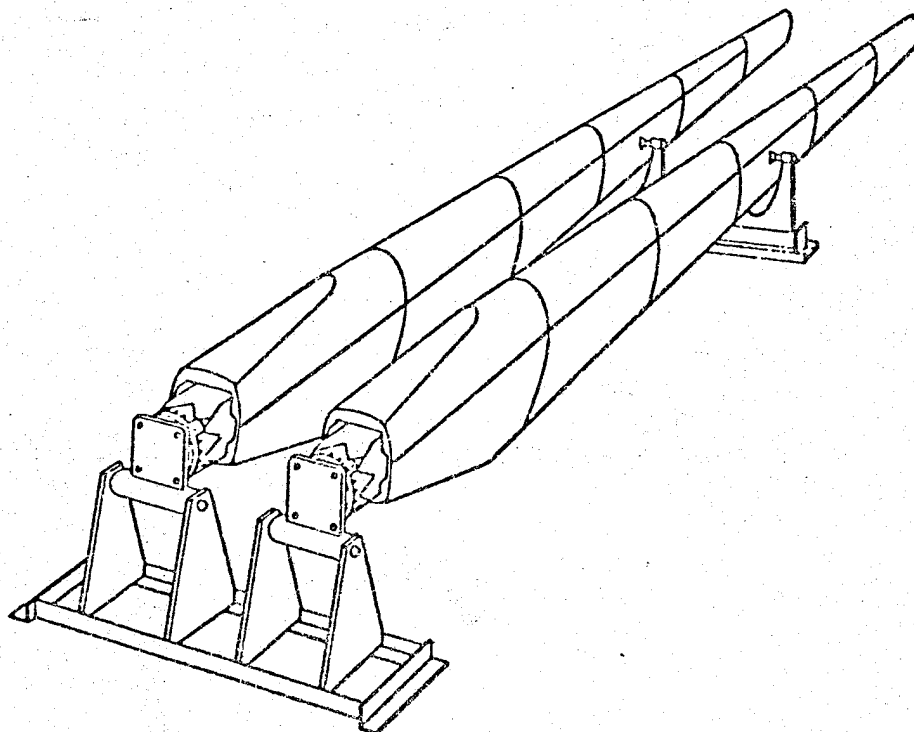
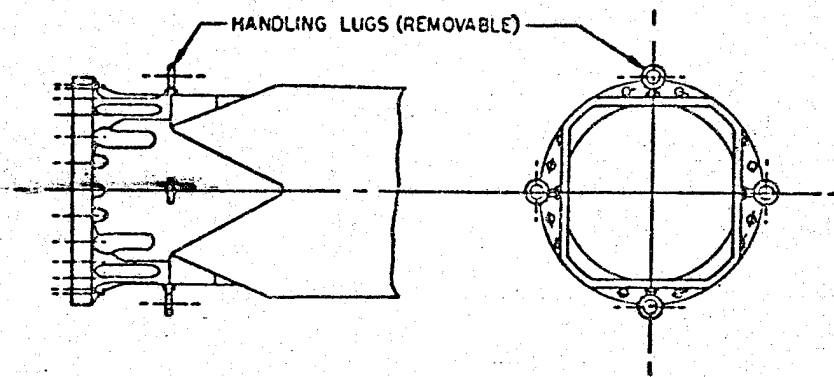FIGURE 33 - SHIPPING FIXTURE ARRANGEMENT
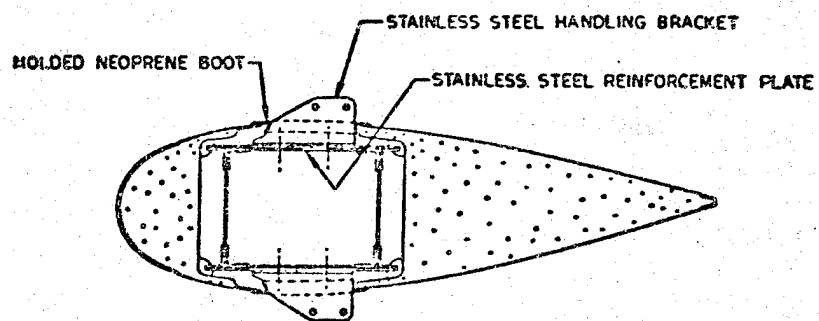
FIGURE 34 - ROOT END HANDLING POINTS

FIGURE 35 - BLADE HANDLING POINTS

FABRICATION OF THE SPAR TEST ELEMENT

To verify the spar design, a root end section of the spar
was designed and fabricated. The test beam element is full
scale and approximately 20 feet long. The root end welding
and attachment is simulated exactly. Shown in Figure 36 is
the design of the root end test beam.

TEST ZONE
APPROX. 72.5 INCHES
FROM ROOT END

7,445 LBS *

240 INCHES

4,227 LBS *

130 INCHES
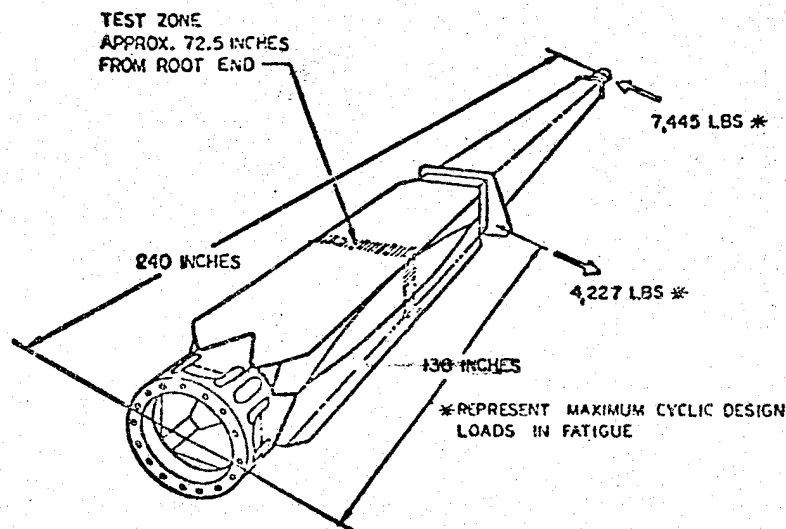
*REPRESENT MAXIMUM CYCLIC DESIGN
LOADS IN FATIGUE

FIGURE 36 - ROOT END TEST BEAM

It is essential for testing a fabricated beam that the bending
shear stresses be simulated in addition to the direct bending
stresses. To accomplish this, we recommend that two opposing
loads be applied to the beam as represented by Figure 36. This
loading design will simulate correctly the combined direct and
shear stresses at station 72.5 and the maximum bending moment
at the root end attachment. Secondly, the design cycle loads
correspond to an "R" value different than that of the planned
test. The recommended loads were calculated on the basis of
equal margins of safety for the test loads and design loads
relative to the Goodman Diagram. The analysis of the test
beam was submitted during the final design review.

Figures 37 through 45 show the beam at various levels of
manufacture.

Figure 37 shows the extension beam assembly and its parts.
This beam is used to extend the test beam to a length of
20 feet to match the NASA test fixture. The parts in the
background are the spot weld and root end arc welding pro-
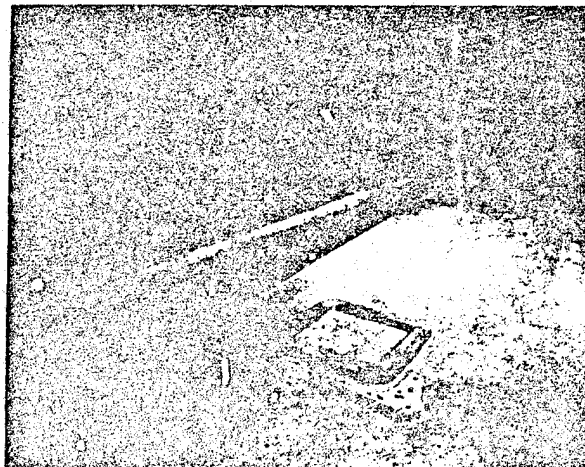cess certification assemblies.

FIGURE 37 - EXTENSION BEAM ASSEMBLY AND PARTS

Figure 38 shows the inboard end of the side plate angle
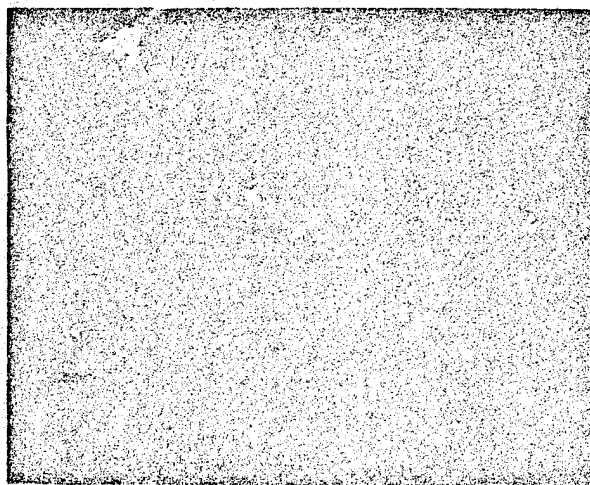assemblies of the spar.

FIGURE 38 - INBOARD END OF SIDE PLATE ANGLE ASSEMBLIES
            OF SPAR

Figure 39 shows the temporary fixture used to assemble the
test beam side and top plate assemblies. It is fabricated
of aluminum and has adjustable fixture blocks to match the
geometry of the top and side assemblies.
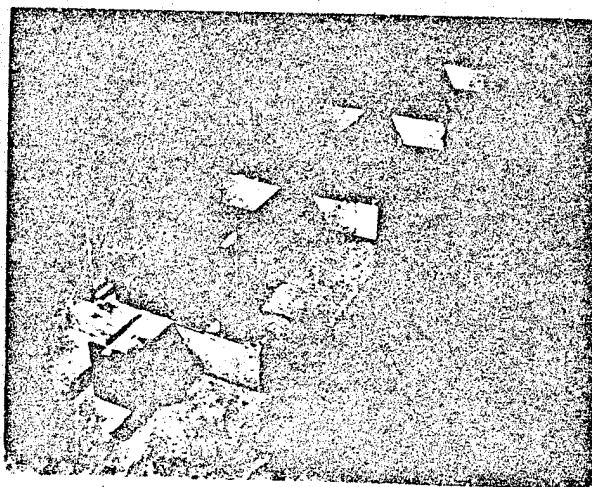

FIGURE 39 - TEMPORARY FIXTURE FOR TEST BEAM SIDE AND
            TOP PLATE ASSEMBLIES

Figure 40 shows the four assemblies completed, ready for
assembling into the box spar.

FIGURE 40 - FOUR ASSEMBLIES READY FOR BOX SPAR

Figure 41 shows the stainless steel spar assembled and the weld preparation prior to attachment of the root end fitting.
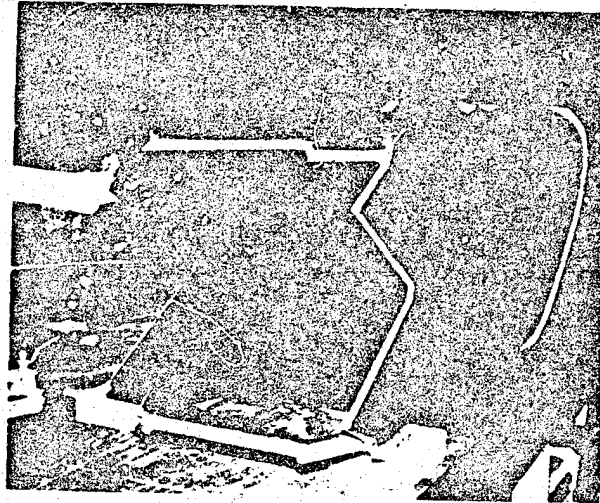

FIGURE 41 - ASSEMBLED BOX SPAR

Figure 42 shows the arc welding configuration of the 4130 carbon steel root end fitting to the stainless steel box spar. The weld was made by preheating the parts during the welding process. The weld has been ground, polished and cold worked on the surface to improve its fatigue re istance.
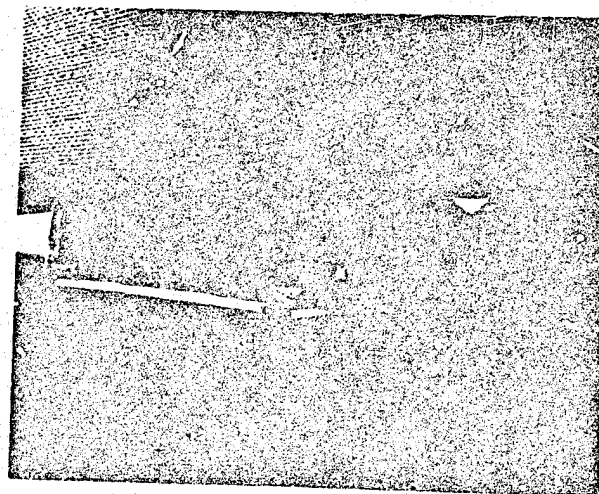

FIGURE 42 - ARC WELDING CONFIGURATION OF 4130 ROOT END
FITTING TO STAINLESS STEEL BOX SPAR

Figure 43 shows the attachment of the carbon steel end
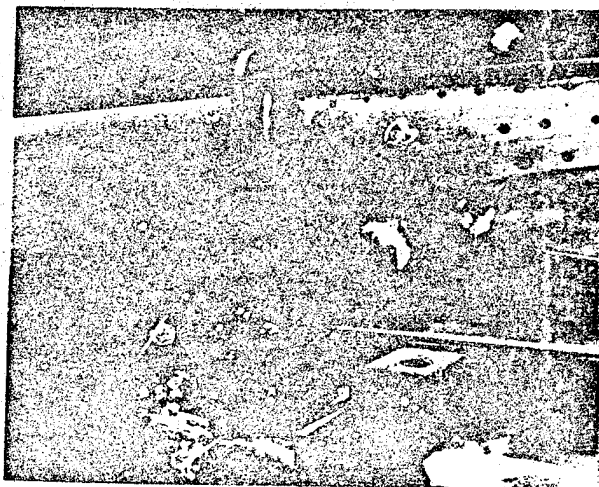extension and the stainless steel spot welded spar.

FIGURE 43 - ATTACHMENT OF CARBON STEEL END EXTENSION
AND STAINLESS STEEL SPOT WELDED SPAR

Figure 44 shows the butt arc weld at the splice between the
.125 inch to .090 inch web plate. The weld has been ground
and polished and then cold worked to provide the high test
allowable. This weld is located at the mid-point of the test
spar, the area of highest stress developed in the spar test.

FIGURE 44 - BUTT ARC WELD SHOWN AT SPLICE BETWEEN .125
TO .090 INCH WEB PLATE

Figure 45 shows another view of the finished test beam
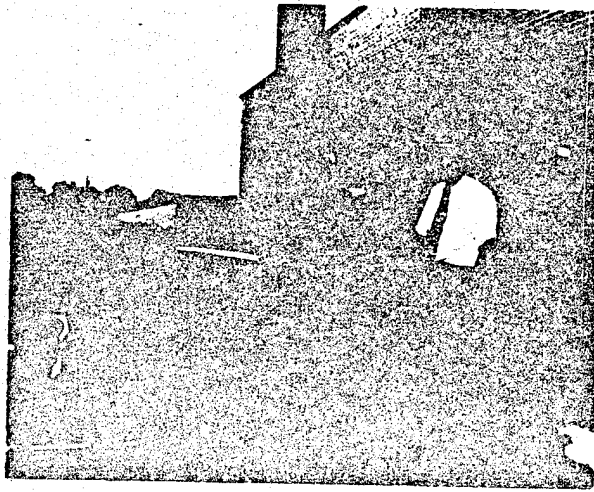(and anchor) before shipping.

FIGURE 45 – TEST BEAM SHOWN BEFORE SHIPPING TO NASA –
LEWIS FOR TESTING

During the manufacture of the test beam, all the processes
required for manufacture of the spar for the 60 foot blade,
were demonstrated and certified. The test beam was shipped
to NASA Lewis for testing by the Government.

CONCLUSIONS

The test beam was fabricated and delivered to the Government.
The fabrication of the test beam certified the processes used
to fabricate the stainless steel spar for the blade. We are
pleased with the product and believe the stainless design used
for the spar has cost and weight advantages for wind turbine
systems 125 ft. diameter and under. The large systems up to
300 ft. diameters would require a fusion welded low alloy car-
bon steel spar. The processes for this spar follows conven-
tional industrial practice with the exception that more complete
and exacting quality control procedures would be used for the
fusion welding processes. The design of the aerodynamic sur-
faces was demonstrated satisfactorily with the system element
test performed during the program. Additional analysis that
was done for application to the large blades indicate that the
Budd approach for fabricaitng and control of the aerodynamic
surfaces is practical and lends itself well for the application.

It was concluded that the Budd blade concept is a viable design
for application to large wind turbine systems. A complete de-
sign for a blade applicable to the MOD-OA system was accomplished
and the blade design satisfies the NASA Lewis Research Center
specifications. The design concept is applicable to various
length blades. Blades less than 60 feet would use a stainless
steel spar and blades of larger size would have a fusion welded
carbon steel spar. The design permits a choice of a wide
selectivity of air-foil shapes and, because of its modular
construction, the design provides producibility and quality in
high volume production.

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15. Supplementary Notes

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16. Abstract

The program task was to design a low-cost wind turbine blade based on a stainless steel – fiberglass-foam Budd blade design concept, to meet NASA Lewis specifications, and to evaluate its principle characteristics, its low cost features, and its advantages and disadvantages. A blade structure was designed and construction methods and materials were selected. A complete blade tooling concepts, various technical and economic analysis, and evaluations of the blade design were performed. A comprehensive fatigue test program was conducted to provide data to verify the design stress allowables. A test specimen of the spar assembly, including the root end attachment was fabricated. This full-scale section, 20 ft. long, will be fatigue tested by NASA.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
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